Introduction and Literature Review

1.1 General

Earth-moving machines are used for engineering projects such as roads, dams, open pit excavation , quarries, trenching, recycling, landscaping and building sites [2]. Among various types of earth-moving machines, Excavator (figure 1.1) is one of the most commonly used machines .

In the past excavator was called dragline because they were using the weight of components to perform excavation jobs and this took very long time and accordingly affects the performance and productivity but nowadays hydraulic Power is used to perform all excavation jobs in hydraulic excavators easily and faster. In this research was taken CAT excavator as a sample to analyses hydraulic circuit component in boom motion (raise high speed, normal raising and normal lowering).

Figure 1.1: Excavator Cat 320

Engines in hydraulic excavators usually just drive hydraulic pumps; there are usually 3 pumps: the two main pumps are for supplying oil at high pressure for the arms, swing motor, track motors, and accessories, and the third is a lower pressure pump for Pilot Control, this circuit used for the control of the spool valves, this allows for a reduced effort required when operating the controls. The two main sections of an excavator are the undercarriage and the house. The undercarriage includes the blade (if fitted), tracks, track frame, and final drives, which have a hydraulic motor and gearing providing the drive to the individual tracks, and the house includes the operator cab, counterweight, engine, fuel and hydraulic oil tanks. The house attaches to the undercarriage by way of a center pin. High pressure oil is supplied to the tracks' hydraulic motors through a hydraulic swivel at the axis of the pin, allowing the machine to slew 360° unhindered [8].

The main boom attaches to the house, and can be one of several different configurations: Most are mono booms: these have no movement apart from straight up and down.

Some others have a knuckle boom which can also move left and right in line with the machine. Another option is a hinge at the base of the boom allowing it to hydraulically pivot up to 180° independent to the house; however, this is generally available only to compact excavator [8].

1.2 Motivation:

One of the major objectives of this study is to analyzed the parameter of hydraulic system component of boom motion. detailed investigation of the hydraulic system component parameters. Furthermore the developed model has to be consolidated afterwards allowing the performance of the hydraulic control system simulations in MATLAB/Simscape/ Simhydraulics.

Another objective of this study is to compare the simulation of boom motion against the motion of boom in the design sheets.

1.3 Scope of work:

This study covered the development of model of the hydraulic system in motion of excavator hydraulic. In addition to that, effects changes in any parameter of hydraulic systems components can be analyzed in a more cost-saving and faster manner with the help of this model.

The purpose of this project is to create a physical model of a hydraulic control system of motion of excavator's power, and to evaluate the results from the simulation. The results will be confirmed by physical readings from the real system.

1.4 Literature Review:

Hydraulics control systems are widely used and a well-known control systems. A number of studies on the performance of hydraulic control systems and its components have been made in recent past. Some of the relevant and significant studies related with the present work are discussed here.

Per-Willy Lazuli and Bjorn Victor Lund [1] presented in a study the results of modeling and simulation of a physical hydrostatic transmission with three different modeling tools; Simulink, SimHydraulics and Simulation X. The aim has been to get the simulations from the different models to be as similar as possible to the two measured pressures and the rotational speed of the load. The Simulation X model gave the best results compared with the measurements. The largest challenge has been to simulate the model in Simulink and to find the frictional losses in the hydraulic motor by performing different tests. The solver in Simulink could not solve the equations and it was difficult to find the tests for finding two of the friction parameters.

Boran Kilic [2] developed a dynamic model of the loader system of a backhoe-loader. Rigid bodies and joints in the loader mechanism and loader hydraulic system components are modeled and analyzed in the same environment using the physical modeling toolboxes, Simhydraulics and Simmechanics, available inside the commercially available simulation software, MATLAB/Simulink. Interaction between the bodies and response of the hydraulic system are obtained by co-operating the mechanical and hydraulic analyses. System variables such as pressure, flow and displacement are measured on a physical machine and then compared with the simulation results. Simulation results are consistent with the measurement results. The main result of this work is the ability to determine the dynamic loads on the joints and attachments of the backhoe-loader. In addition to that, prototyping time and costs can be highly reduced by implementing this model in the design process.

Weinan Cao, etc [3] analyzed the necessity of research on Model and Simulation of ocean wave power generation platform of hydraulic lifting system. Aiming to the working condition and mechanism of self-elevating power-generating platform, the paper provided overall designing scheme and analyzed the working principle of the system .By modeling and simulation of hydraulic system based on AME Sim software to set different parameter values for analysis of dynamic characteristics and stability of the system, which can help to find out the factors that influence the dynamic characteristics. This founding has a certain significant guiding meaning for the parameter optimization design of the jack up platform of hydraulic lifting system.

DESCRIPTION OF THE SYSTEM

The controlled movement of parts or a controlled application of force is a common requirement in the industries. These operations are performed mainly by using electrical machines or diesel, petrol and steam engines as a prime mover. These prime movers can provide various movements to the objects by using some mechanical attachments like screw jack, lever, rack and pinions etc. However, these are not the only prime movers. The enclosed fluids (liquids and gases) can also be used as prime movers to provide controlled motion and force to the objects or substances. The specially designed enclosed fluid systems can provide both linear as well as rotary motion. The high magnitude controlled force can also be applied by using these systems. This kind of enclosed fluid based systems using pressurized incompressible liquids as transmission media are called as hydraulic systems [4].

A hydrostatic system uses fluid pressure to transmit power. Hydrostatics deals with the mechanics of still fluids and uses the theory of equilibrium conditions in fluid. The system creates high pressure, and through a transmission line and a control element, this pressure drives an actuator (linear or rotational). The pump used in hydrostatic systems is a positive displacement pump. The relative spatial position of this pump is arbitrary but should not be very large due to losses (must be less than 50 m). An example of pure hydrostatics is the transfer of force in hydraulics [5].

The hydrostatic system works on the principle of Pascal's law which says that the pressure in an enclosed fluid is uniform in all the directions. Pascal's law is illustrated in figure 2.1. The force given by fluid is given by the multiplication of pressure and area of cross section. As the pressure is same in all the direction, the smaller piston feels a smaller force and a large piston feels a large force. Therefore, a large force can be generated with smaller force input by using hydraulic systems [5].

Hydrodynamic systems use fluid motion to transmit power. Power is transmitted by the kinetic energy of the fluid. Hydrodynamics deals with the mechanics of moving fluid and uses flow theory. The pump used in hydrodynamic systems is a non-positive displacement pump. The relative spatial position of the prime mover (e.g., turbine) fixed. An example of pure hydrodynamics is the conversion of flow energy in turbines. In hydroelectric power plant in oil hydraulics, we deal mostly with the fluid working in a confined system, that is, a hydrostatic system [5].

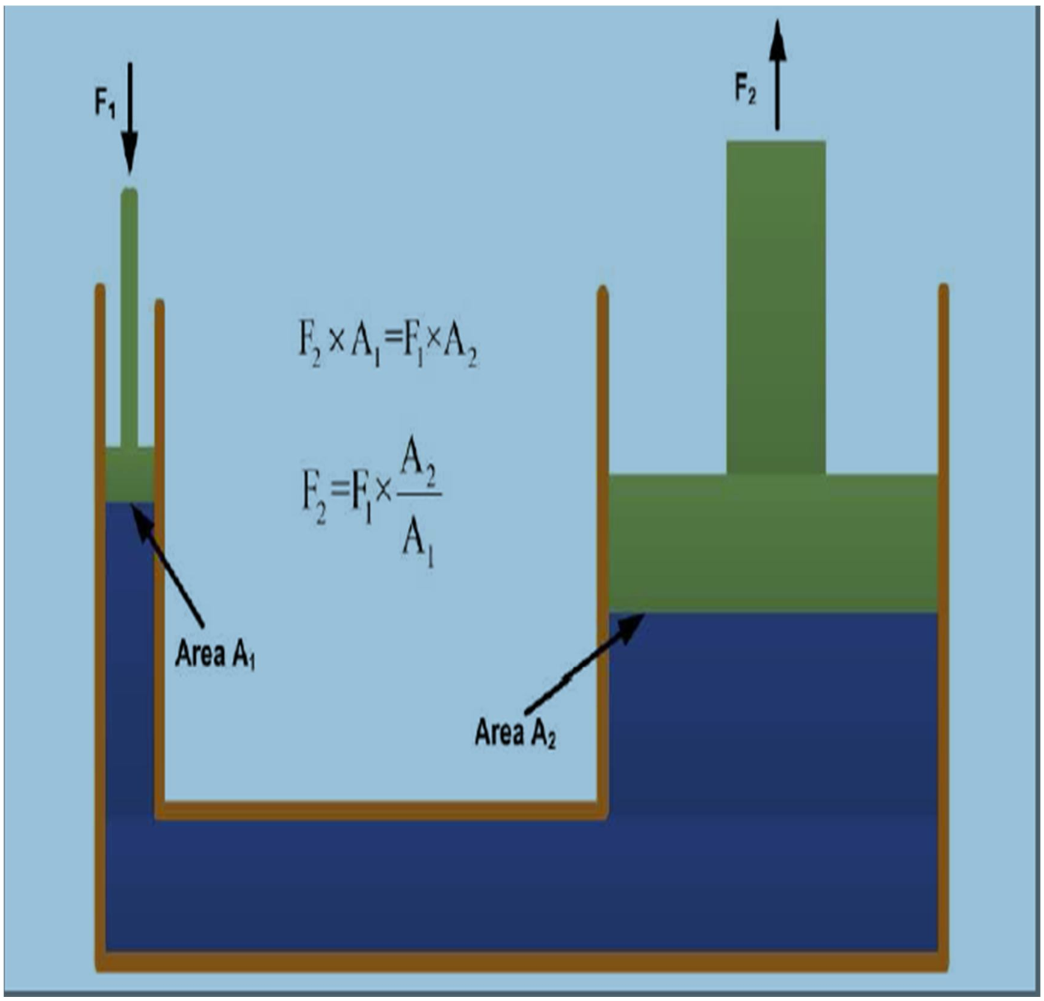

Figure 2.1: Principle of hydrostatic system

2.2 Basic Hydraulic System Components

In hydraulic control systems, electrical energy or thermal energy is used to drive an electrical motor or combustion engine and then mechanical energy is generated. This mechanical energy is then converted into hydraulic energy by means of hydraulic pump, processed in open or closed loops to drive linear or rotary actuators (cylinders or motors) by converting it back into mechanical energy to do the required work (see figure 2.2) .

As an example figure 2.3 shows a simple circuit of a hydraulic system with basic components Functions of the components in the hydraulic circuit are as follows:

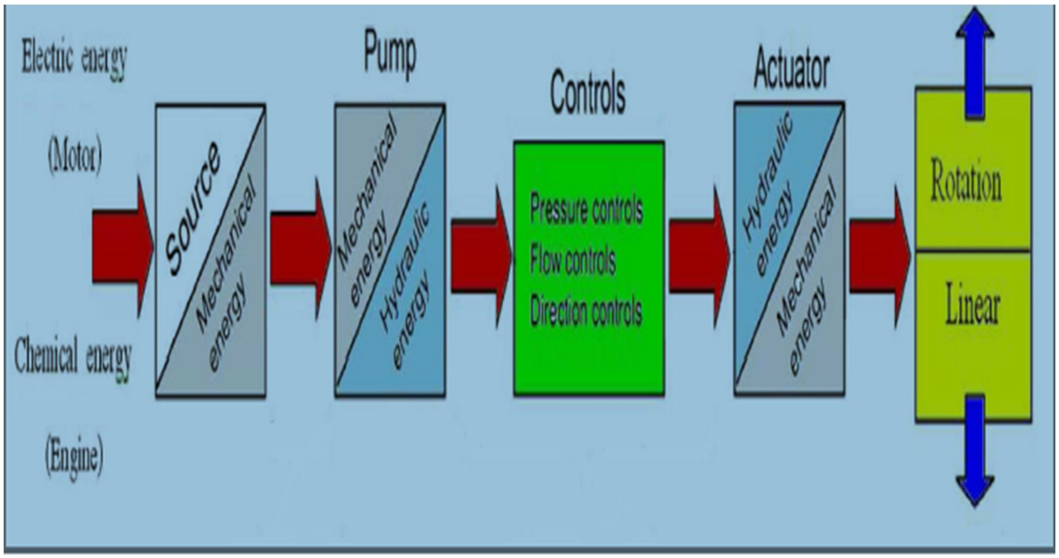

Figure 2.2: Energy conversion in hydraulic system

2.1.1. External power supply (motor) is used to provide the pump with the required mechanical power.

2.1.2. The hydraulic pump is used to force the fluid from the reservoir to rest of the hydraulic circuit by converting mechanical energy into hydraulic energy.

2.1.3. Reservoir is used to hold the hydraulic liquid, usually hydraulic oil.

2.1.4. Piping system carries the hydraulic oil through various components of the hydraulic system.

2.1.5. Filters are used to remove any foreign particles so as keep the fluid system clean and efficient, as well as avoid damage to the actuator and valves.

2.1.6. Pressure regulator regulates (i.e., maintains) the required level of pressure in the hydraulic fluid

2.1.7. Valves are used to control the direction, pressure and flow rate of a fluid flowing through the circuit

2.1.8. The hydraulic actuator is a device used to convert the fluid power into mechanical power to do useful work. The actuator may be of the linear type (e.g., hydraulic cylinder) or rotary type (e.g., hydraulic motor) to provide linear or rotary motion, respective Cylinder movement is controlled by a three-position change over a control valve.

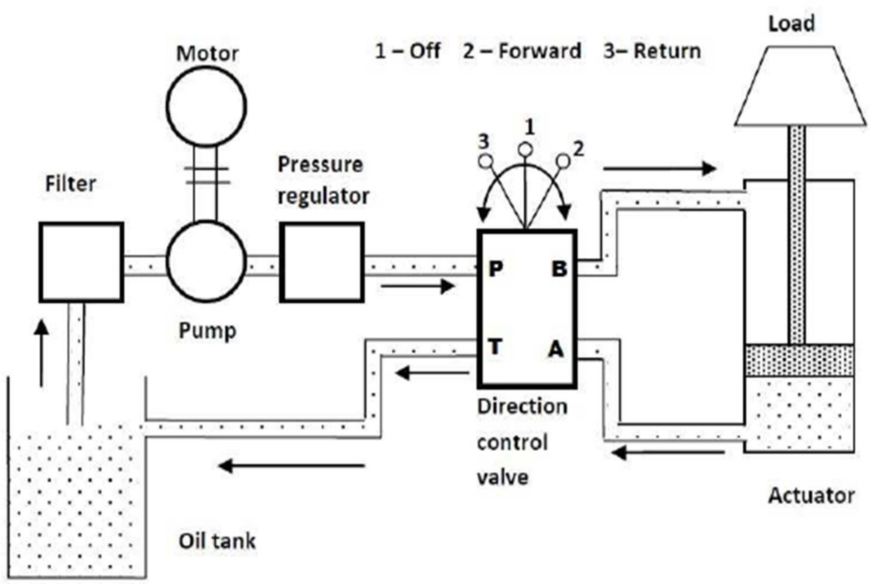

Figure2.3: simple hydraulic circuit

- When the piston of the valve is changed to upper position, the pressure port P is connected to port A and tank port T is connected to port B to allow oil in the upper side of the piston to return back to the tank, thus the load is raised.
- When the position of the valve is changed to lower position, the pressure port P is connected to port B and tank port T is connected to port A to allow oil in the upper side of the piston to return back to the tank, thus the load is lowered.
- When the valve is at center position, it locks the fluid into the cylinder, there by holding it in position.

In industry, a machine designer conveys the design of hydraulic systems using a circuit diagram. Figure 2.4 shows the components of the hydraulic system using symbols. The working fluid, which is the hydraulic oil, is stored in a reservoir.

When the electric motor is switched ON, it runs a positive displacement pump that draws hydraulic oil through a filter and delivers at high pressure. The pressurized oil passes through the regulating valve and does work on actuator. Oil from the other end of the actuator goes back to the tank via return line. To and fro motion of the cylinder is controlled using directional control valve [5].

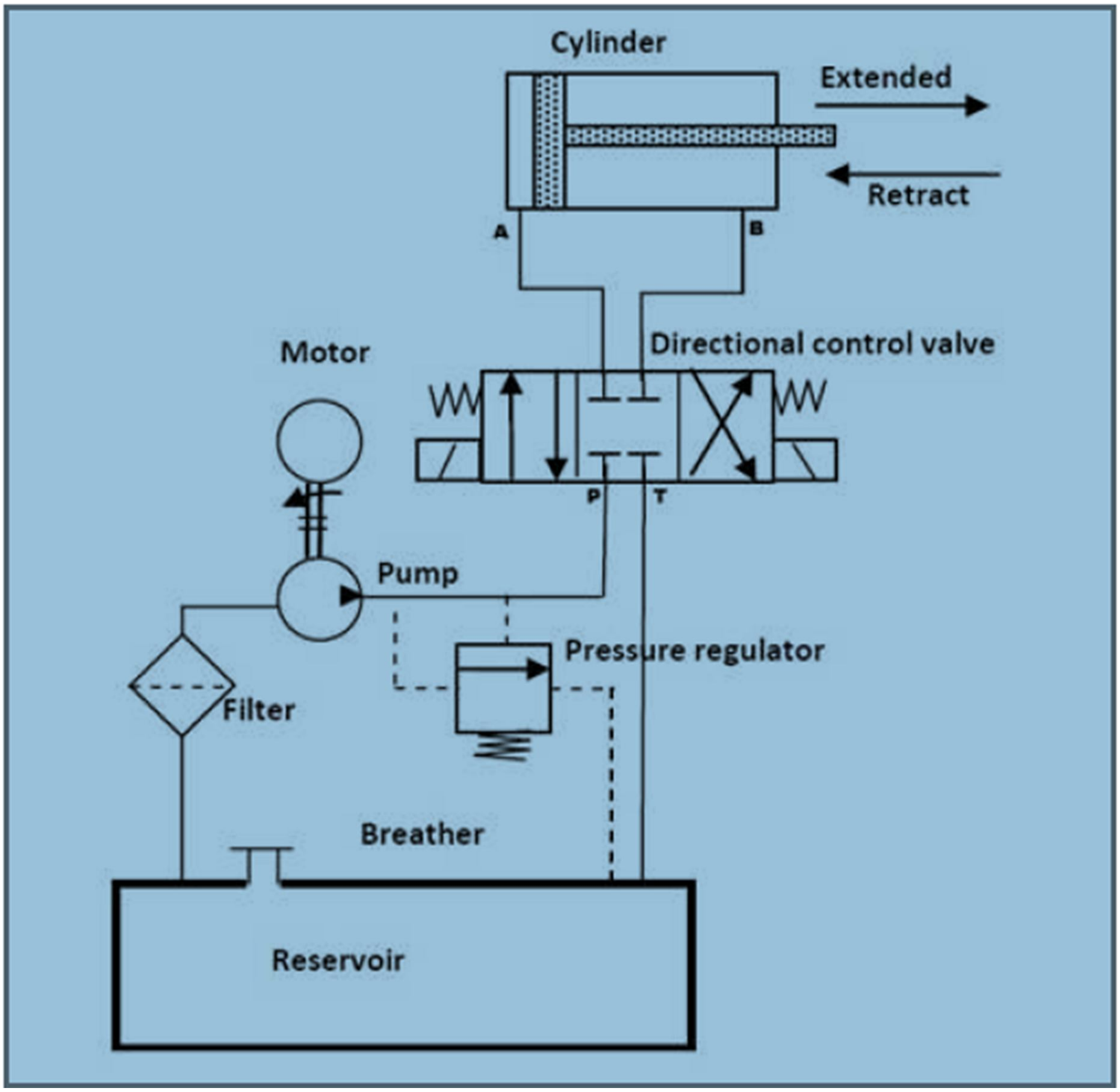

Figure 2.4: Hydraulic system components shown in symbols

The hydraulic system discussed above can be broken down into four main divisions that are analogous to the four main divisions in an electrical system.

1. The power device parallels the electrical generating station.
2. The control valves parallel the switches, resistors, timers, pressure switches relays, etc.
3. The pipes in which the fluid power flows parallel the electrical lines.
4. The fluid power actuator (whether it is a linear or rotating, cylinder or hydraulic motor) parallels electrical motors and the solenoid.

So in general, hydraulic control system consists of the following components shown in block diagram as illustrated in figure 2.5

1. The drive block which consists of a source of mechanical energy (electrical motors, combustion engines) connected to a Hydraulic pump.
2. Control block which consists of valves so as to control:
   - Direction and it is called directional control valves.
   - Pressure and it is called pressure control valves.
   - Flow and it is called flow control valves.
3. The output block or Actuators to do the useful work, whether it is linear (cylinder) or rotary (motor).
4. The machine block represents the load or the mechanical work to be done[6].

2.3 Main Components Of Hydraulic System

2.3.1 Control Valve

2.3.2 Travel & Swing Motors

2.3.3 Main pumps

2.3.4 Hydraulic Tank.

2.3.5 Boom Cylinders

.2.3.6 Stick Cylinder

2.3.7 Bucket Cylinder

Figure 2.6 shows the Main Components of hydraulic system . in excavator

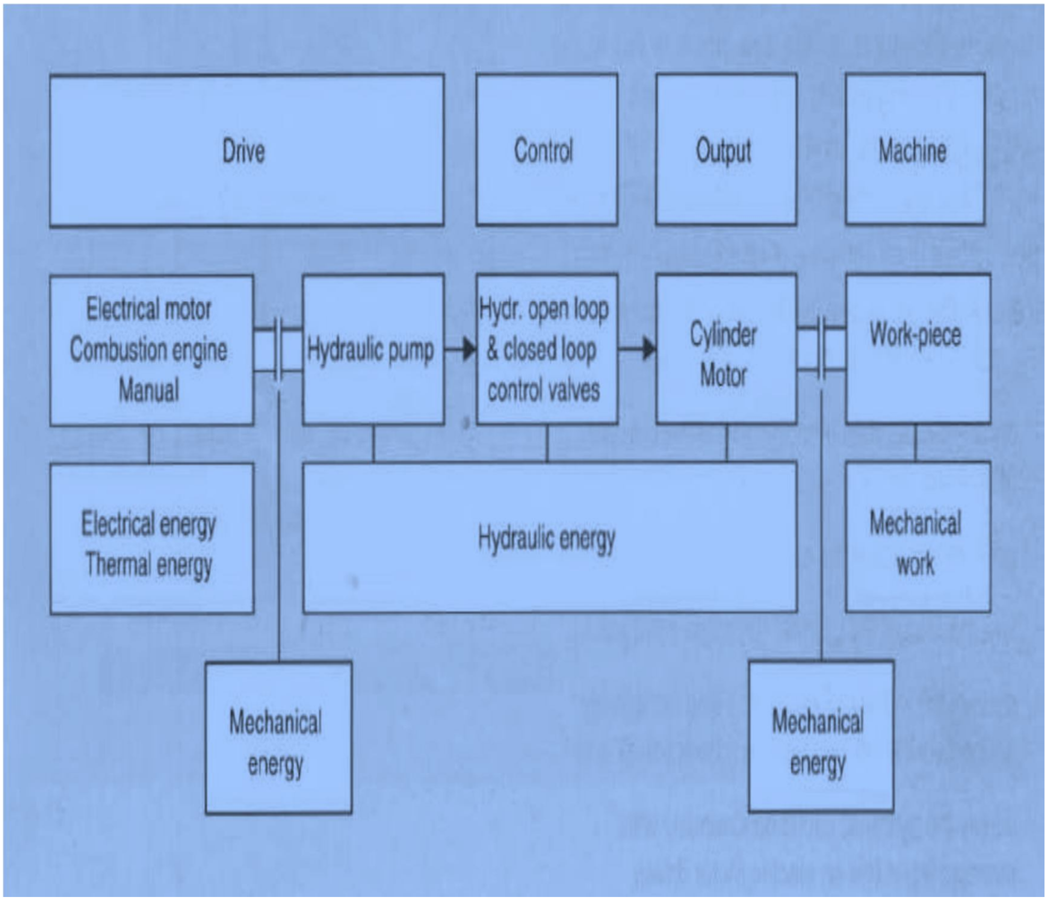

Figure2.5: block diagram of hydraulic system [6].

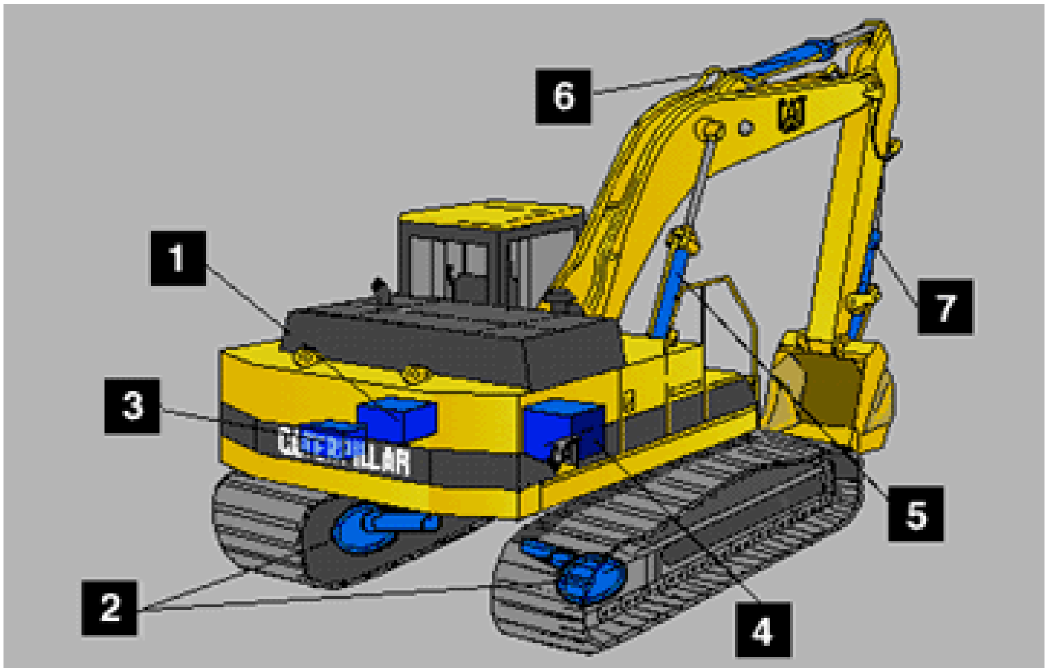

Figure 2.6 Hydraulic Main Components

2.4 Pilot Control Valves

Figure 2.7a & figure 2.7b shows the control of hydraulic system excavator boom, stick, and bucket motion. Also figure 2.8 shows the monitor of the hydraulic system.

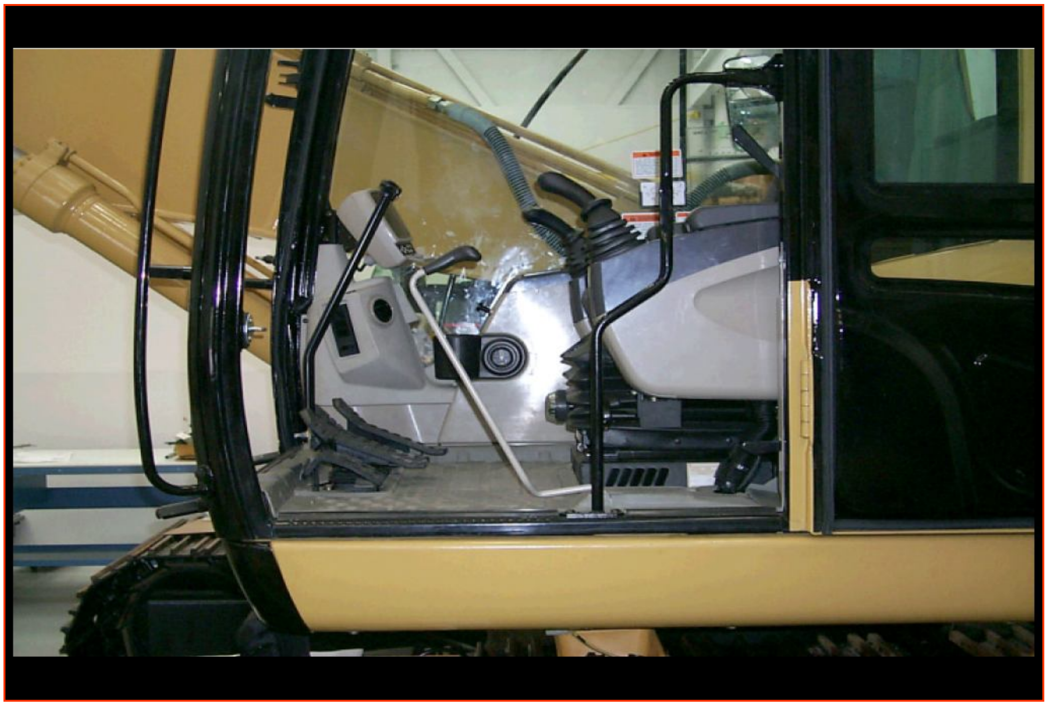

Figure 2.7 a Pilot Control Valves

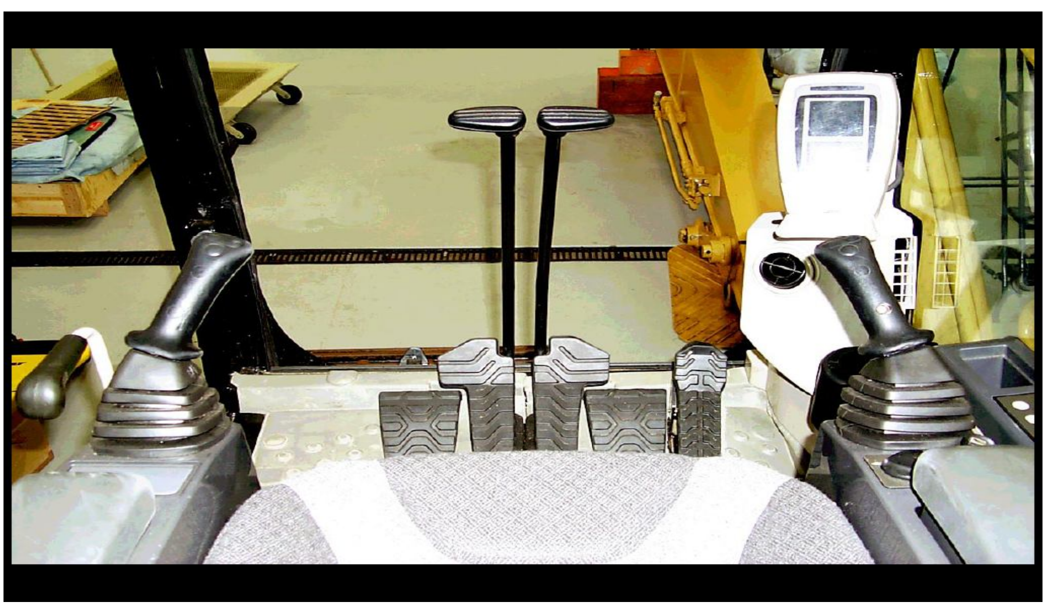

Figure 2.7 b Pilot Control Valves

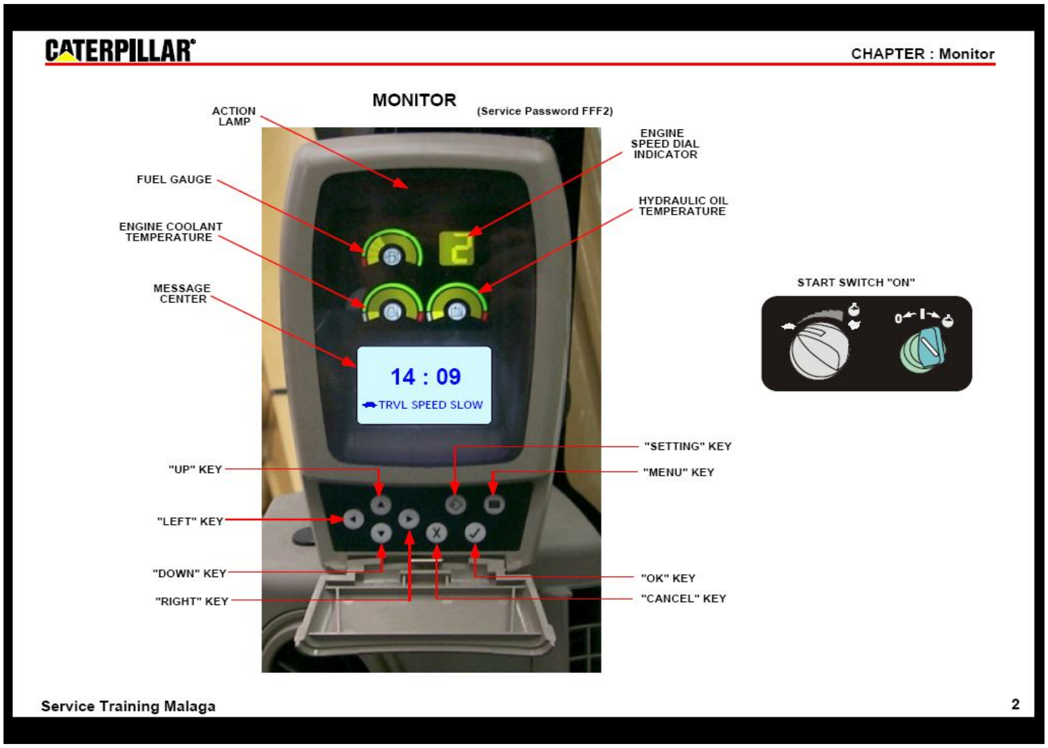

Figure 2.8 The Monitor

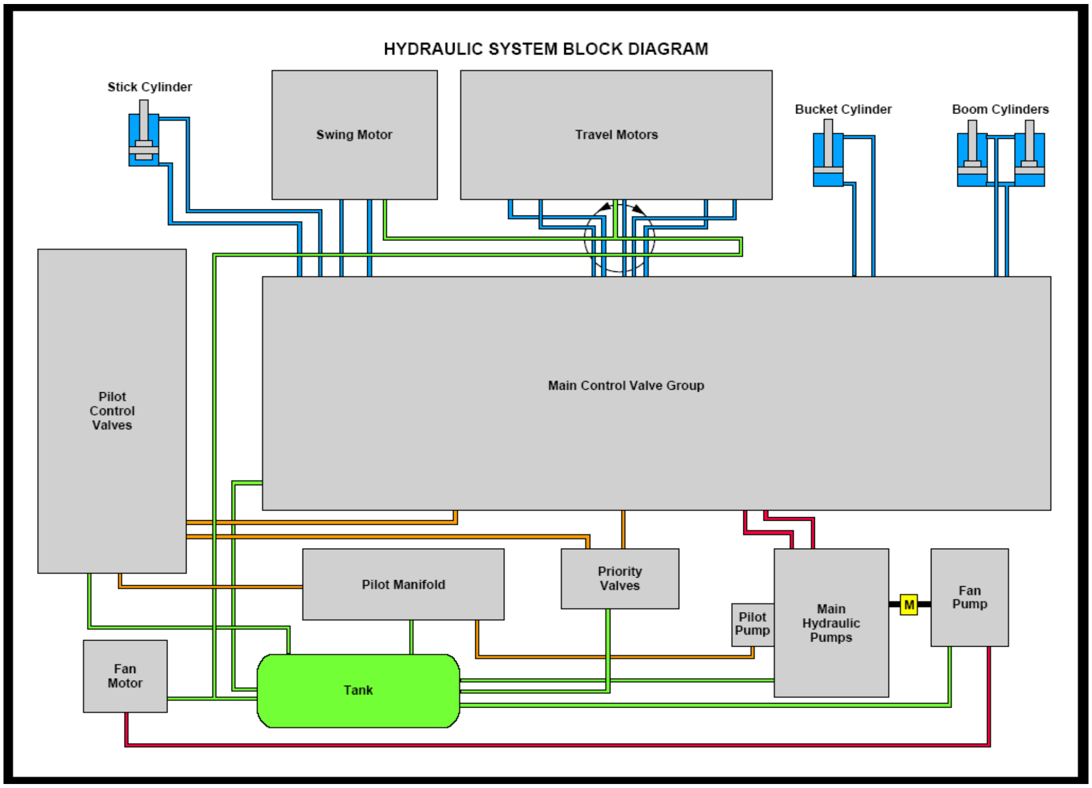

Figure 2.9: Hydraulic block system diagram[7]

2.5 Boom Hydraulic System and Operation

2.5.1 Boom Raise (High Speed)

According to figure 2.10, that is represents Hydraulic schematic for BOOM RAISE (high speed), the detail components as:

- (1) Boom cylinders
- (2) Line (oil flow from boom cylinder rod end)
- (3) Line (oil flow to boom cylinder head end)
- (4) Valve
- (5) Boom drift reduction valve
- (6) Return line
- (7) Port
- (8) Parallel feeder passage
- (9) Return passage
- (10) Line
- (11) Main control valve
- (12) Passage
- (13) Check valve
- (14) Load check valve
- (15) Port
- (16) Boom II control valve
- (17) Parallel feeder passage
- (18) Return passage
- (19) Boom I control valve
- (20) Port
- (21) Pilot line
- (22) Pilot control valve (boom and bucket)
- (23) Pilot line
- (24) Pilot line
- (25) Pilot line
- (26) Pressure reducing valve for boom priority
- (27) Left pump
- (28) Right pump
- (29) Pilot pump
- (33) Spring
- (37) Spring

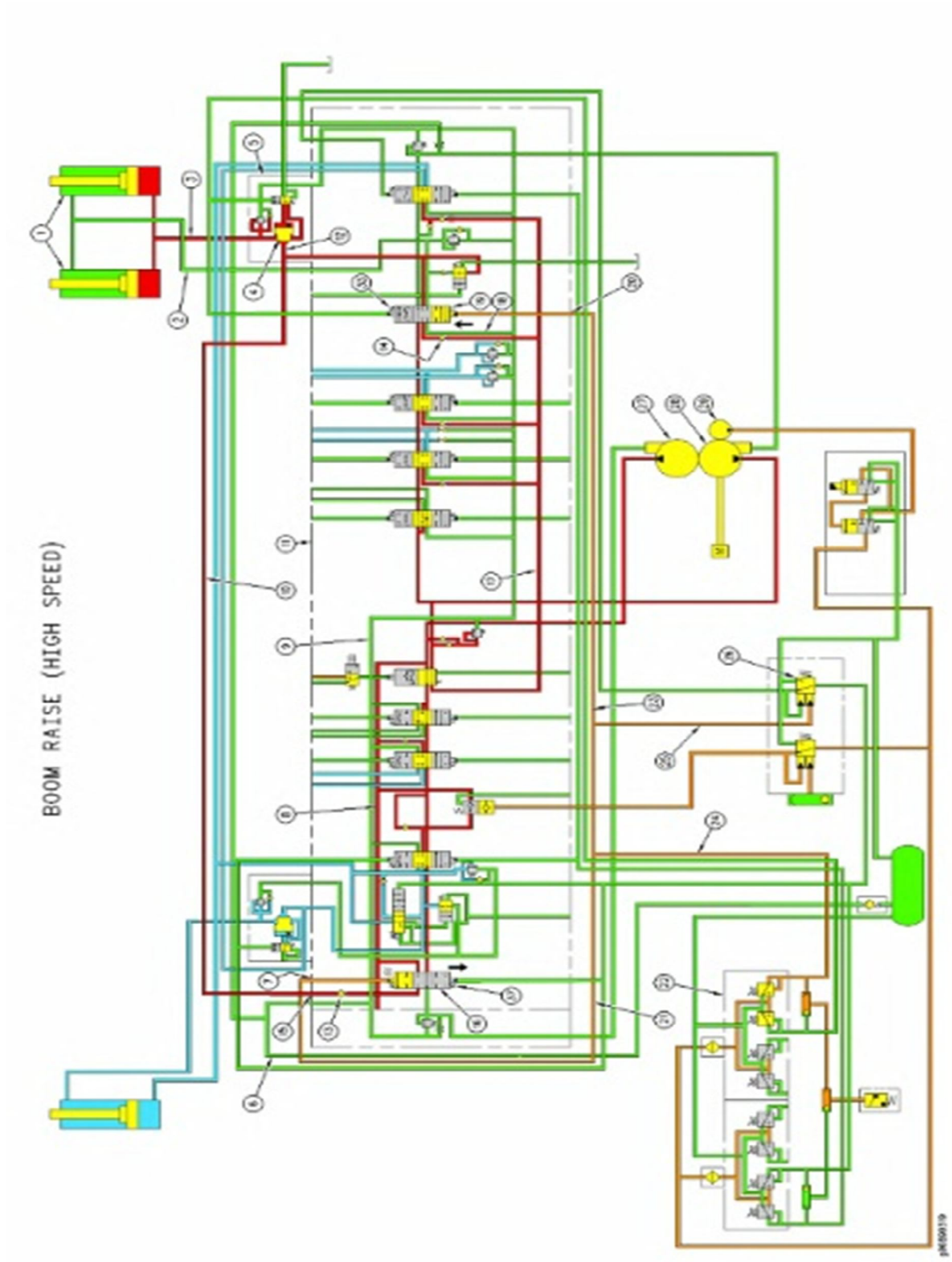

Figure 2.10: Hydraulic Schematic For Boom Raise (high speed)[7]

A BOOM RAISE operation at high speed is accomplished when the oil delivery from both left pump (27) and right pump (28) is supplied to the head end of boom cylinders (1). Boom I control valve (19) and boom II control valve (16) operate during the high speed operation. A BOOM RAISE operation at low speed is accomplished when the oil delivery from only right pump (28) is supplied to the head end of boom cylinders (1). During the low speed operation, boom I control valve (19) operates alone.

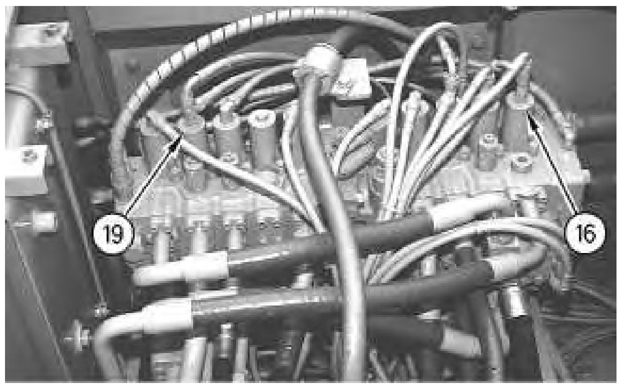

Figure 2.11: Main control valve compartment

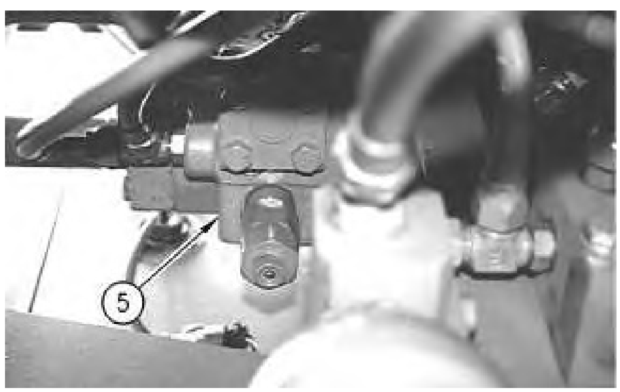

Figure 2.12: Boom drift reduction valve (bottom view)

The oil delivery from right pump (28) flows through parallel feeder passage (17) in main control valve (11) to boom I control valve (19). The oil delivery from left pump (27) flows through parallel feeder passage (8) in main control valve (11) to boom II control valve (16).

When the joystick for the boom is moved to the full BOOM RAISE position, the pilot oil flows from pilot control valve (22) through pilot line (24). The pilot oil flow then divides into two flow paths. Part of the pilot oil flows through pilot line (21) to port (7) of main control valve (11). The remainder of the pilot oil flows through pilot line (23) to port (20) of the main control valve.

A portion of the oil in pilot line (23) also flows through pilot line (25) to the pressure reducing valve for boom priority (26). During a combined operation of BOOM RAISE and STICK IN, the pilot oil flow to the pressure reducing valve for boom priority (26) causes the boom circuit to receive oil flow priority. This allows the boom to raise at a high speed.

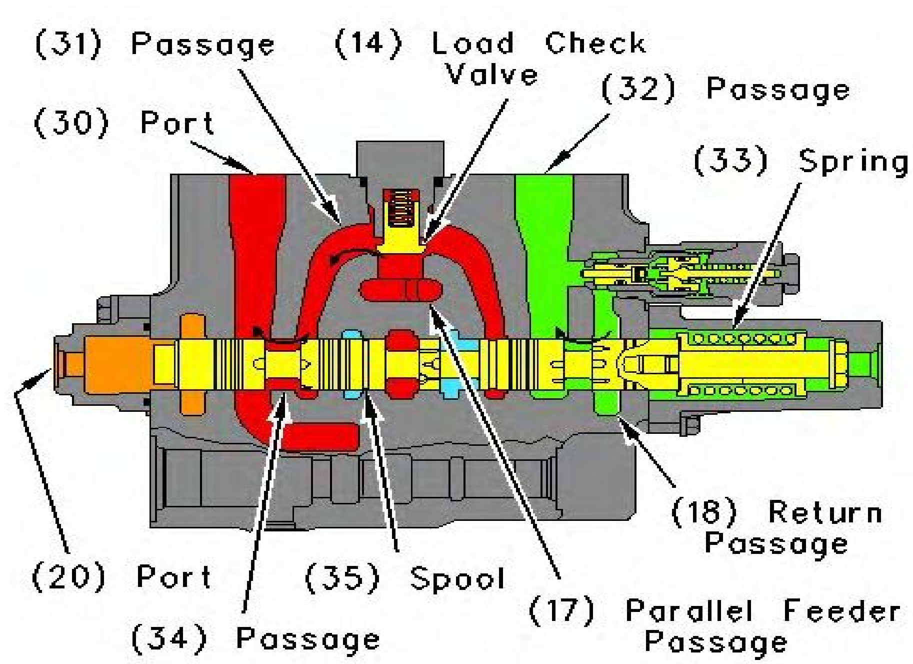

Figure 2.13: Boom I Control Valve (Boom Raise Position)

We can see from figure 2.13, that,

- (14) Load check valve
- (17) Parallel feeder passage
- (18) Return passage
- (20) Port
- (30) Port

- (31) Passage
- (32) Passage
- (33) Spring
- (34) Passage
- (35) Spool

The pilot oil flow from port (20) shifts spool (35) of boom I control valve (19) against the force of spring (33). The oil delivery from the right pump in parallel feeder passage (17) flows through load check valve (14), passage (31), passage (34) and port (30) to boom drift reduction valve (5). The oil delivery from the right pump shifts valve (4) in boom drift reduction valve (5) to the right. The oil delivery from the right pump then flows through line (3) to the head end of boom cylinders (1) [7].

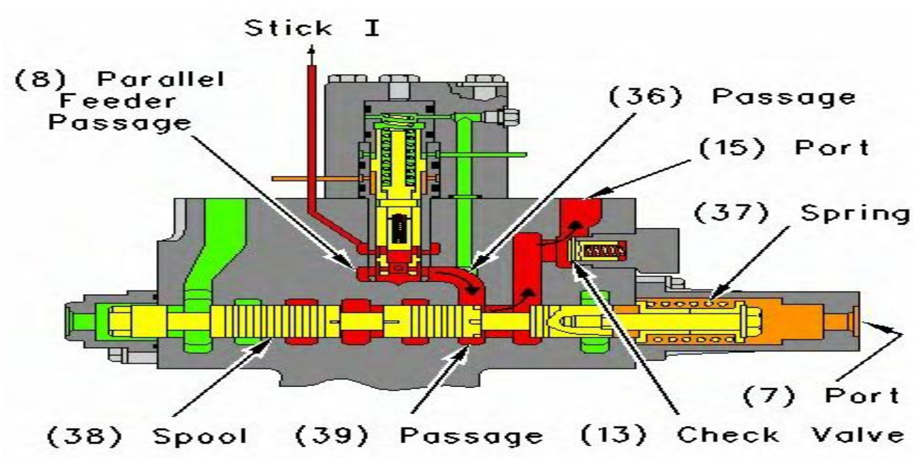

Figure 2.14: Boom II Control Valve (Boom Raise Position)

We can see from the figure 2.14 that.

- (7) Port
- (8) Parallel feeder passage
- (13) Check valve
- (15) Port
- (36) Passage
- (37) Spring
- (38) Spool
- (39) Passage

The pilot oil flow in port (7) of boom II control valve (16) shifts spool (38) against the force of spring (37). The oil delivery from the left pump in parallel feeder passage (8) now flows through passage (36), passage (39), check valve (13) and flows out of port (15) to line (10). The oil delivery from the left pump combines with the oil delivery from the right pump at boom drift reduction valve (5). The combined pump oil flows through passage (12) and line (3) to the head end of boom cylinders (1).

Note: The swing priority valve does not affect the boom II control valve.

Return oil from the rod end of boom cylinders (1) flows through line (2) to boom I control valve (19). The oil then flows through passage (32), return passage (18), return passage (9) and return line (6) to the hydraulic tank[7].

2.5.2 Boom Raise (Low Speed)

According to figure 2.15, that is represents Hydraulic schematic for BOOM RAISE (low speed), the detail components as:

(1) Boom cylinders

(2) Line (oil flow to boom cylinder rod end)

(3) Line (oil flow from boom cylinder head end)

(4) Valve

(5) Boom drift reduction valve

(14) Load check valve

(16) Boom II control valve

(17) Parallel feeder passage

(18) Return passage

(19) Boom I control valve

(22) Pilot control valve (boom and bucket)

(27) Left pump

(28) Right pump

(29) Pilot pump

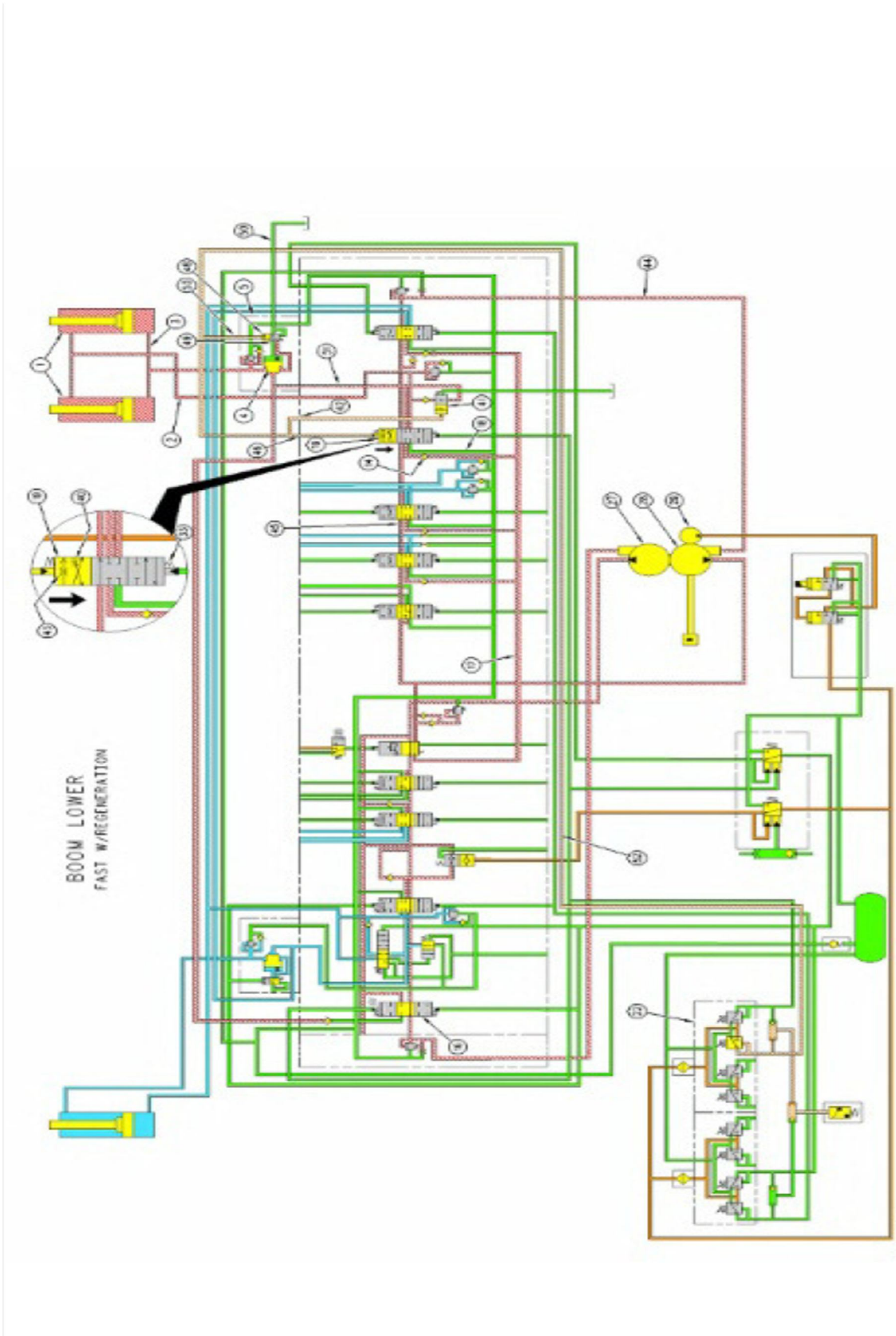

Figure (2.15): Hydraulic Schematic For Boom Raise (Low speed) [7]

(33) Spring

(40) Orifice

(41) Boom regeneration valve

(42) Port

(43) Orifice

(44) Negative flow control line

(45) Center bypass passage

(46) Port

(48) Valve

(49) Passage

(50) Drain line

(51) Passage

(52) Pilot line

(53) Pilot line

When the joystick for the boom is moved less than half of the travel distance for BOOM RAISE, low pilot oil pressure is supplied to boom I control valve (19) and boom II control valve (16) .

When the boom is raised at a low speed, boom I control valve (19) opens and boom II control valve (16) remains closed. The force of spring (33) in boom I control valve (19) is less than the force of spring (37) in boom II control valve (16). Because of the low pilot oil pressure, boom I control valve (19) will open and boom II control valve (16) will remain closed.

The oil delivery from right pump (28) now flows to the head end of boom cylinders (1). Without the oil delivery from left pump (27), the cylinder rod movement slows down when the boom is raised. The low speed operation of the boom is performed [7] .

2.5.3 Boom Lower Operation

During a BOOM LOWER operation, the oil delivery from only right pump (28) is supplied to boom cylinders (1) through boom I control valve (19). Boom I control valve (19) operates alone. Boom II control valve (16) is not operational in the BOOM LOWER operation.

The BOOM LOWER operation contains a regeneration circuit. When the joystick for the boom is moved to the BOOM LOWER position, orifice (43) in boom I control valve (19) and boom regeneration valve (41) are operational in the boom hydraulic circuit. The return oil flow from the head end of boom cylinders (1) flows through boom regeneration valve (41) to the rod end of the boom cylinders.

When the joystick for the boom is moved to the BOOM LOWER position, pilot oil from pilot control valve (22) flows through pilot line (52). The pilot oil flow then divides into three flow paths. Part of the pilot oil flows through port (46) to boom I control valve (19). Part of the pilot oil flows through port (42) to boom regeneration valve (41). The remainder of the pilot oil flows through pilot line (53) of boom drift reduction valve (5).

Since the pilot oil pressure has caused the spool in boom I control valve (19) to shift against the force of spring (33), the oil delivery from the right pump that flows through center bypass passage (45) is restricted by orifice (43). The negative flow control pressure in negative flow control line (44) decreases. The right pump upstrokes because of the negative flow control operation.

We can see from figure (2.16), that

(14) Load check valve

(17) Parallel feeder passage

(18) Return passage

(30) Port

(32) Port

(33) Spring

(35) Spool

(42) Orifice

(45) Orifice

(46) Port

(49) Passage

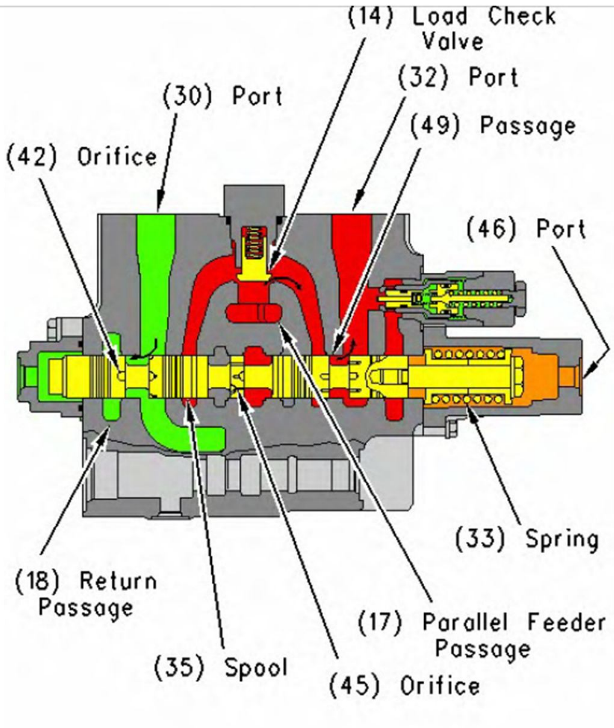

Figure (2.16): Boom I Control Valve (Boom Lower Position)

The pilot oil flow from port (46) shifts spool (35) in boom I control valve (19) against the force of spring (33). The oil delivery from the right pump in parallel feeder passage (17) flows through load check valve (14), passage (49) and port (32). The oil delivery from the right pump then flows through line (2) to the rod end of boom cylinders (1).

The return oil from the head end of boom cylinders (1) flows through line (3) into boom drift reduction valve (5). Since valve (48) is shifted by the pilot pressure from pilot line (53), passage

(49) is open to drain line (50). The return oil pressure shifts valve (4) to the right. The return oil in line (3) enters passage (51).

A portion of the return oil flows into port (30) of boom I control valve (19). The return oil flow is restricted by orifice (40). The return oil pressure in passage (51) increases. Most of the return oil flows through boom regeneration valve (41). The return oil is now supplied to the rod end of the boom cylinders through line (2).

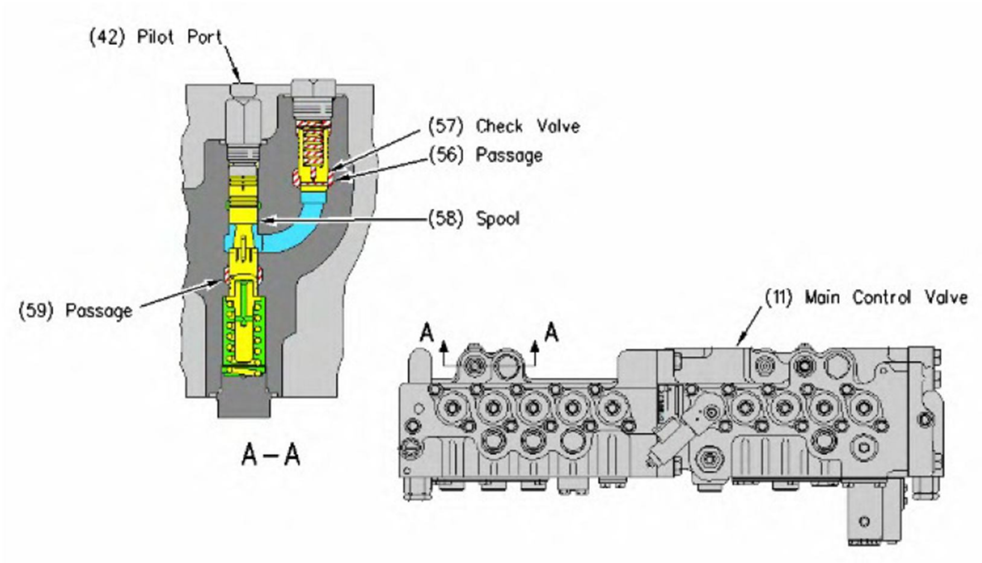

Figure (2.17): Boom Regeneration Valve (Slow Boom Down)

The figure 2.17 shows the flowing components

(11) Main control valve

(42) Pilot port

(56) Passage

(57) Check valve

(58) Spool (boom regeneration valve)

(59) Passage

The boom hydraulic circuit contains a regeneration circuit. This regeneration circuit allows the return oil from the head end of the boom cylinders to be supplied to the rod end of the boom cylinders during the BOOM LOWER operation.

When the joystick for the boom is moved to the BOOM LOWER position, pilot oil flow from the pilot control valve (boom and bucket) enters pilot port (42). Spool (58) in the boom regeneration valve shifts downward. The return oil from the head end of the boom cylinders flows through passage (59) and through the throttling slots on the spool for the boom regeneration valve to check valve (57). Check valve (57) opens and the return oil flows through passage (56). The return oil from the head end of the boom cylinders in passage (56) combines with the oil delivery from the right pump. This combined oil now flows to the rod end of the boom cylinders.

The oil delivery from only the right pump is used for the BOOM LOWER operation. Since the boom regeneration valve supplies return oil from the head end to the rod end of the boom cylinders, more efficient use of the oil delivery from the right pump is achieved during a BOOM LOWER operation [7].

SIMULATION MODEL

A physical model of the hydraulic control system of power hydraulic of boom motion has been created with MATLAB/ Simscape/SimHydraulics toolbox. The simulation results from the model will be compared against require values from the real system and the error will be found if there was error .

SimHydraulics is a more powerful tool in modeling hydraulic systems .Instead of deriving, programming and solving dynamics equations of hydraulic system components, SimHydraulics provides more detailed component blocks for modeling hydraulic components in the Simulink environment. In SimHydraulics library, there are over 50 different blocks which include linear and rotary actuators, pumps, valves, pipelines. One of the most important advantages of this toolbox is that a SimHydraulics model can be connected to a mechanical system for a multidomain simulation. Moreover, a SimHydraulics system model closely resembles the hydraulic schematic, which lets the user to understand and analyze the model much more efficiently.

The SimHydraulics models is shown in figure 3.1 and figure 3.2. figure 3.1 shows boom motion raising high speed with two pump, and figure 3.2 shows boom motion normal raising and normal lowering with one pump. The two models is constructed with the same components as the actual system. All the parameters of the Simhydraulics blocks were extracted from components datasheets, some of them can be found directly and others must be calculated.

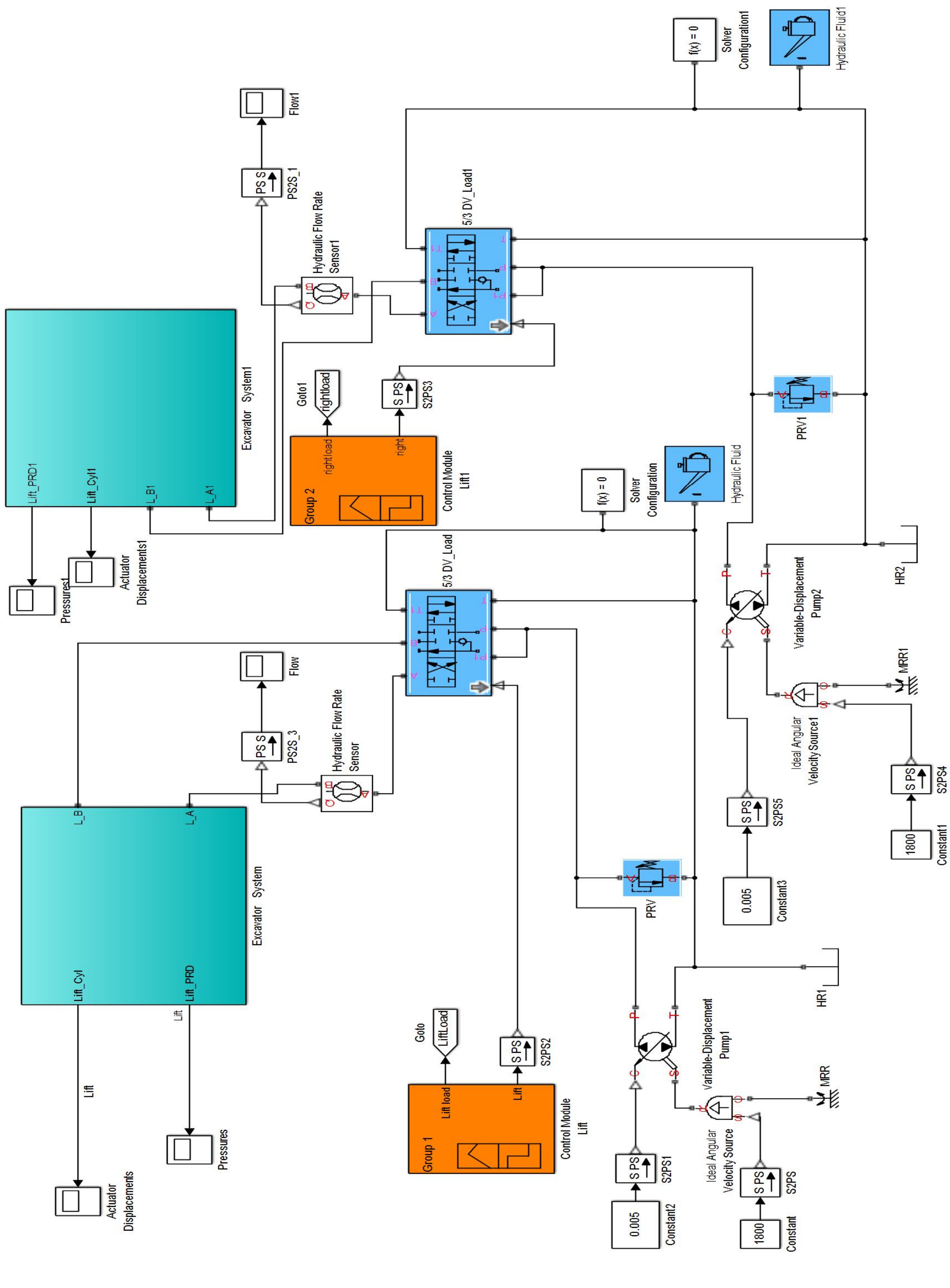

Figure 3.1 SimHydraulics Model with two pump

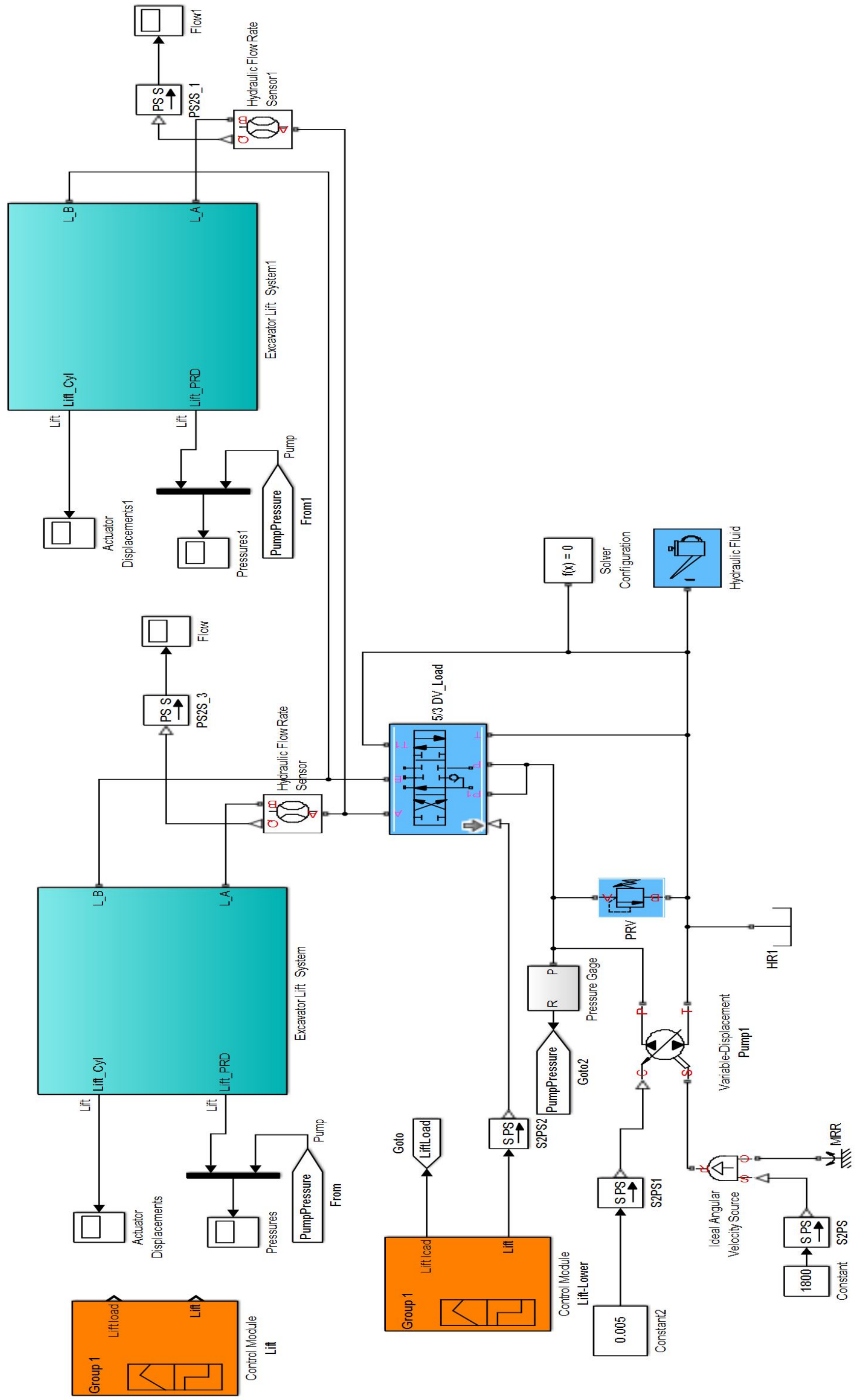

Figure 3.2: SimHydraulics Model with one pump

3.2 Oil Tank Model

The main oil tank and the overhead tank were modeled using Hydraulic Reference (see figure 3.2). The Hydraulic Reference block represents a connection to atmospheric pressure. Hydraulic conserving ports of all the blocks that are referenced to atmosphere (for example, suction ports of hydraulic pumps, or return ports of valves, cylinders, pipelines, if they are considered directly connected to atmosphere) must be connected to a Hydraulic Reference block. The block has one hydraulic conserving port [9].

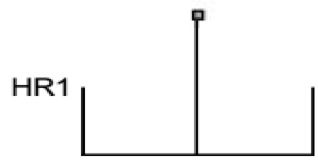

Figure 3.3: Hydraulic Reference

3.3 Electrical motors model

The two electrical motors are modeled. Each model is constructed with the Ideal Angular Velocity Source and the mechanical reference blocks. The Angular velocity source is connected to the mechanical reference and to the Electric Control signal, which is constant block connected to Simulink converter block (see figure 3.3).

This block receives a physical input signal, which is required rotational speed of the motor, from port S and output it to port R which is input to the hydraulic pump.

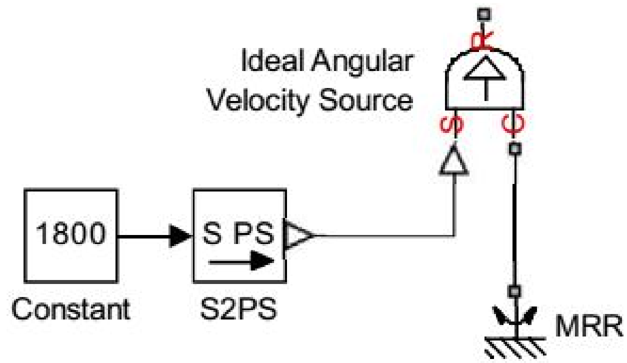

Figure 3.4: Electrical Motor

3.4 Hydraulic Pumps Model

The two hydraulic pumps were modeled using the Variable-Displacement Pump block. The block represents a The Variable-Displacement Pump block represents a variable-displacement bidirectional pump of any type as a data-sheet-based model. The pump delivery is proportional to the control signal provided through the physical signal port C. The pump efficiency is determined based on volumetric and total efficiencies, nominal pressure, and angular velocity. All these parameters are generally provided in the data sheets or catalogs. Port S is connected to the electrical motor, Port T to the main oil tank, and Port P is connected to the pressure line, (see figure 3.5 and figure 3.1).

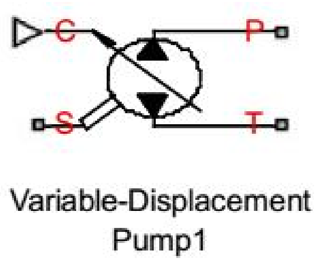

Figure 3.5: Hydraulic Pump block

The key parameters required for this block are pump displacement, volumetric and total efficiencies, nominal pressure, stroke and angular velocity (see figure 3.6). All these parameters are generally provided in the data sheets or catalogs.

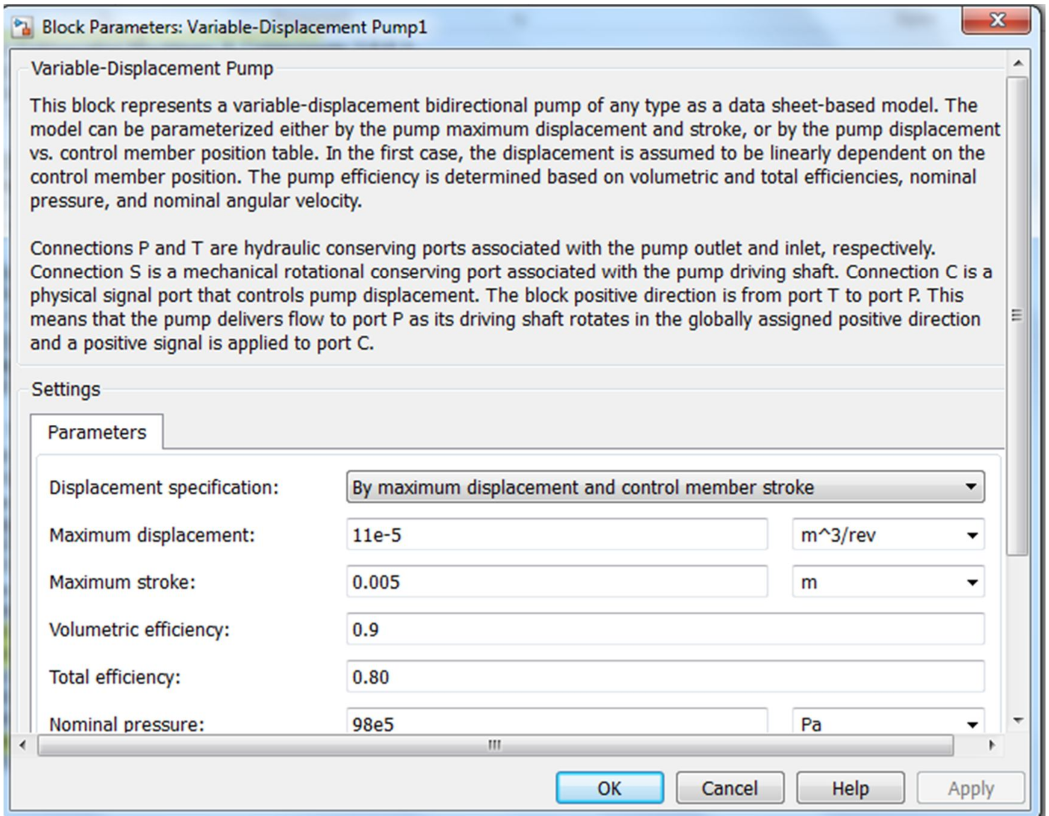

Figure 3.6: parameters in pump block

Then these parameters are used in the calculations of the following equations:

$$T = D * P/\eta_{mech} \quad \rightarrow \quad (3.1)$$

$$q = D * \omega - k_{leak} * P \quad \rightarrow \quad (3.2)$$

$$D = \begin{cases} (D_{max}/x_{max}).\,x \\ D(x) \end{cases} \quad \rightarrow \quad (3.3)$$

$$k_{leak} = \frac{k_{HP}}{v.\rho} \quad \rightarrow \quad (3.4)$$

$$\frac{k_{HP}}{v.\rho}k_{HP} = \frac{D_{max} * \omega_{nom}(1-\eta_V)v_{nom} * \rho_{nom}}{P_{nom}} \quad \rightarrow \ (3.5)$$

$$P = P_P - P_T \quad \rightarrow \ (3.6)$$

The leakage flow is determined based on the assumption that it is linearly proportional to the pressure differential across the pump and can be computed by using the Hagen-Poiseuille formula

$$P = \frac{128\mu l}{\pi D^2} q_{leak} = \frac{\mu}{k_{HP}} q_{leak} \quad \rightarrow \ (3.7)$$

The pump mechanical efficiency is not usually available in data sheets, therefore it is determined from the total and volumetric efficiencies by assuming that the hydraulic efficiency is negligibly small

$$\eta_{mech} = \frac{\eta_{total}}{\eta_V} \quad \rightarrow \ (3.8)$$

The block positive direction is from port T to port P. This means that the pump transfers fluid from T to P as its driving shaft S rotates in the globally assigned positive direction and a positive signal is applied to port C.

3.5 PRV Model

The two PRVs in the system were modeled using the pressure relief valve block which represents a hydraulic pressure relief valve as a data-sheet-based model (see figure 3.7).

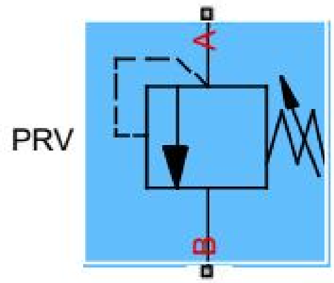

Figure 3.7: Pressure Relief Valve Block

Typical dependency between the valve passage area A and the pressure differential p across the valve is shown in figure 3.8.

The valve remains closed while pressure at the valve inlet is lower than the valve preset pressure. When the preset pressure is reached, the value control member (spool, ball, poppet, etc.) is forced off its seat, thus creating a passage between the inlet and outlet. Some fluid is diverted to a tank through this orifice, thus reducing the pressure at the inlet. If this flow rate is not enough and pressure continues to rise, the area is further increased until the control member reaches its maximum. At this moment, the maximum flow rate is passing through the valve. The value of a maximum flow rate and the pressure increase over the preset level to pass this flow rate are generally provided in the catalogs. The pressure increase over the preset level is frequently referred to as valve steady state error, or regulation range. The valve maximum area and regulation range are the key parameters of the block [9].

In addition to the maximum area, the leakage area is also required to characterize the valve. The main purpose of the parameter is not to account for possible leakage, even though this is also important, but to maintain numerical integrity of the circuit by preventing a portion of the system from getting isolated after the valve is completely closed. An isolated or "hanging" part of the system could affect computational efficiency and even cause failure of computation. Theoretically, the parameter can be set to zero, but it is not recommended. Figure 3.8 shows parameters used in pressure relief valve block [9].

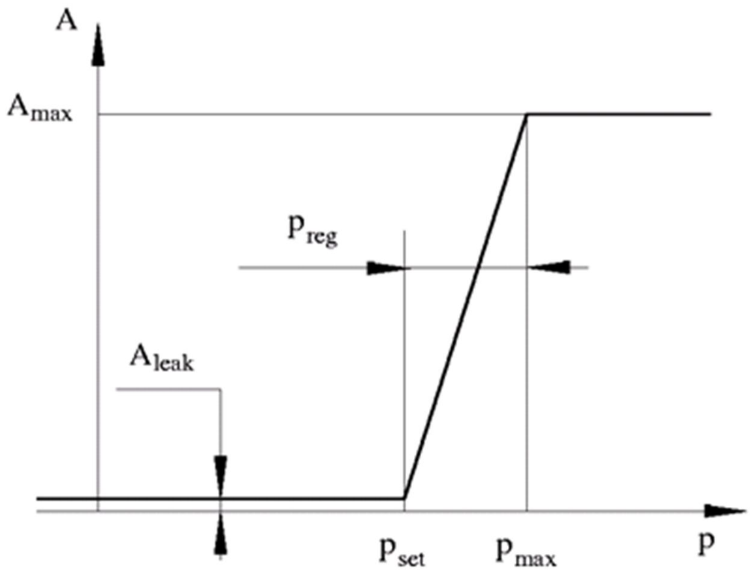

Figure 3.8: Relation Between Passage Area and Pressure Differential Across The Pressure Relief Valve

The flow rate is determined according to the following equations:

$$q = C_d A(p) \sqrt{\frac{2}{\rho}} . \frac{p}{(p^2 + p_{cr}^2)^{1/4}} \qquad \rightarrow (3.9)$$

$$p = p_A + p_B \qquad \rightarrow (3.10)$$

$$p_{cr} = \frac{\rho}{2\left(\frac{Re_{cr} * v}{C_D * D_H}\right)^2} \qquad \rightarrow (3.11)$$

$$A(p) = \begin{cases} A_{leak} \; for \; p \leq \; p_{set} \\ A_{leak} + k(p - p_{set}) \; for \; p_{set} < p < p_{max} \\ A_{max} \; for \; p \geq \; p_{max} \end{cases}$$

$$k = \frac{A_{max} - A_{leak}}{p_{reg}} \qquad \rightarrow (3.12)$$

$$D_H = \sqrt{\frac{4A(p)}{\pi}} \qquad \rightarrow (3.13)$$

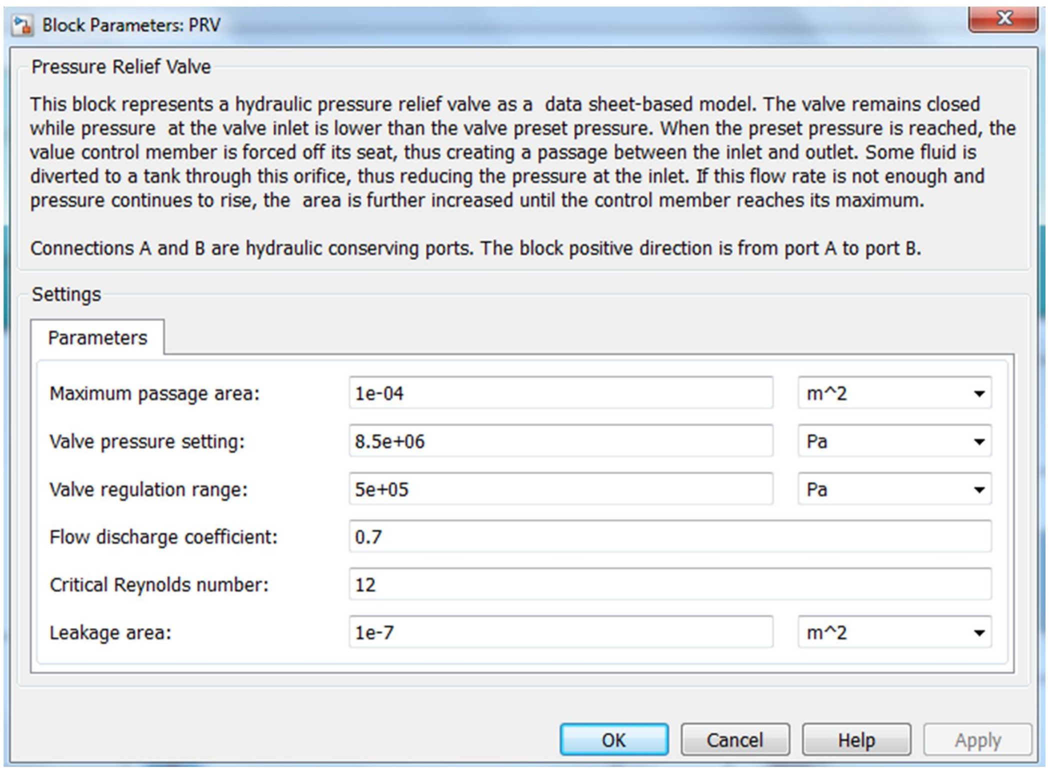

Figure 3.9: Parameters in Pressure Relief Valve Block

3.6 DCV model

The 5-Way Directional Valve A block simulates a configuration of hydraulic continuous 5-way directional valve with pump port P, and p1 two return ports T and T1, and two actuator ports A, and B. Use the valve for applications with two actuators, each being controlled by a valve of this type. When both valves are in neutral position, the pump is unloaded. If any of the valves is shifted from neutral, the diverting line is cut off and the respective actuator is fed at fuel pump pressure. See figure (3.10).

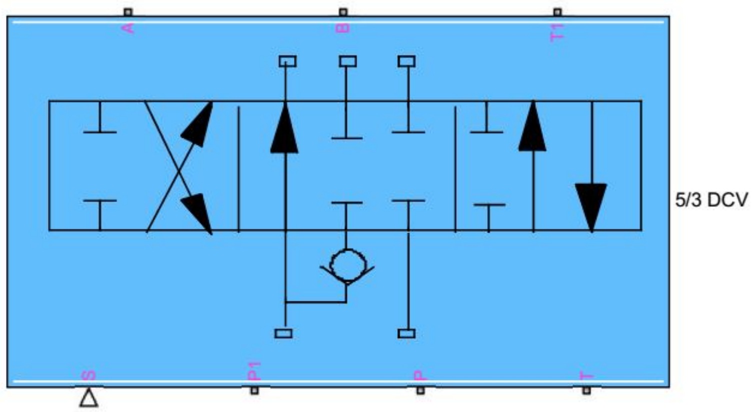

Figure 3.10: 5/3 Directional Valve

The physical signal port connection (S), which controls the spool position. The block is built of five Variable Orifice blocks, connected as shown in the following diagram in figure (3.11). Figure 3.11 shows the parameters used in the direction control valve block.

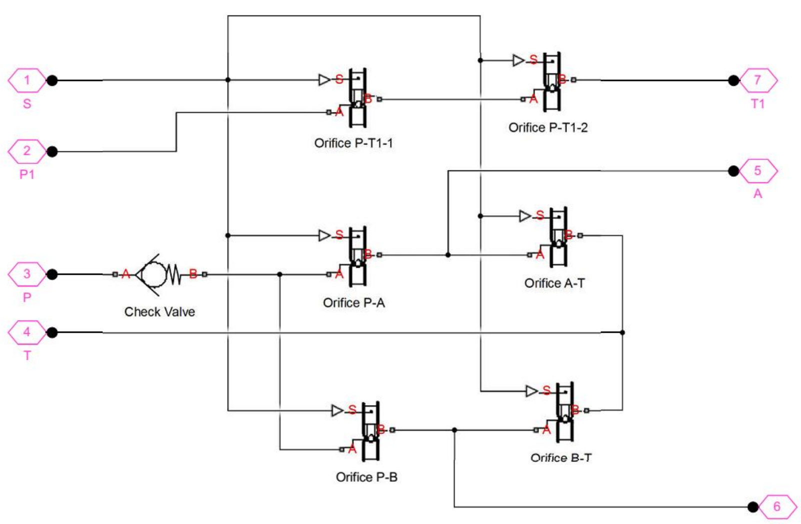

Figure (3.11): block built of 5/3 DCV

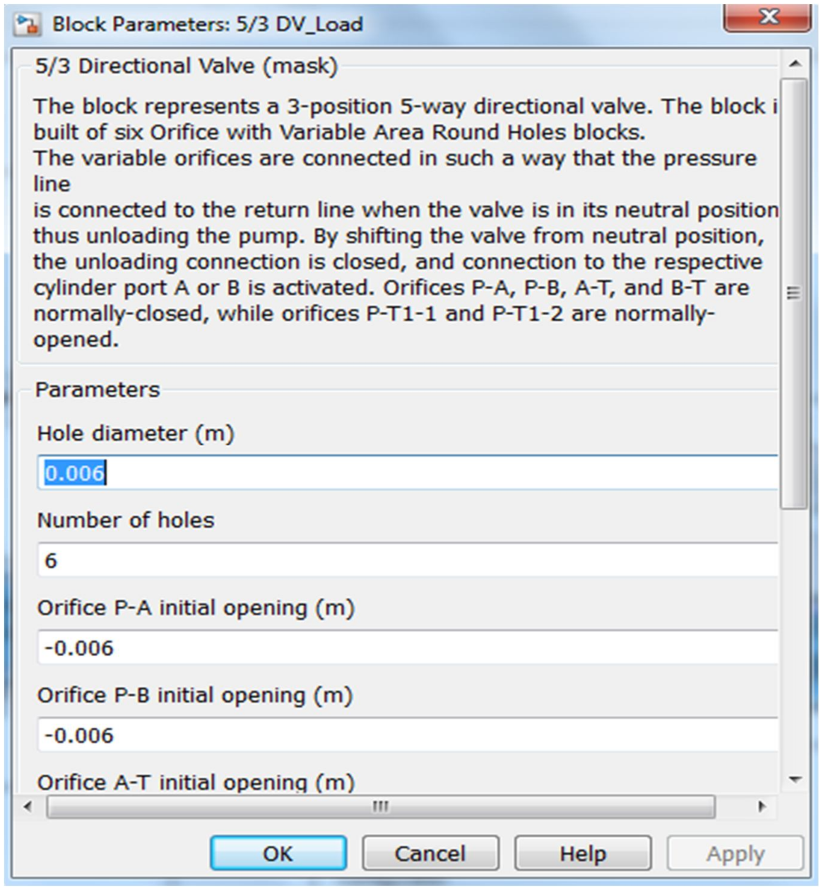

Figure 3.12: Parameters in Directional Valve Block

3.7 Excavator System Model

The excavator subsystem is consists of linear cylinder actuator, and load which is the task to do raise.

3.7.1 The Hydraulic Cylinder Model

The hydraulic cylinder was modeled using the Double-Acting Hydraulic Cylinder block. The Double-Acting Hydraulic Cylinder block models a device that converts hydraulic energy into mechanical energy in the form of translational motion. Hydraulic fluid pumped under pressure into one of the two cylinder chambers forces the piston to move and exert force on the cylinder rod. Double-acting cylinders transfer force and motion in both directions. Parameters used in the Hydraulic cylinder block are shown in figure 3.12 [9].

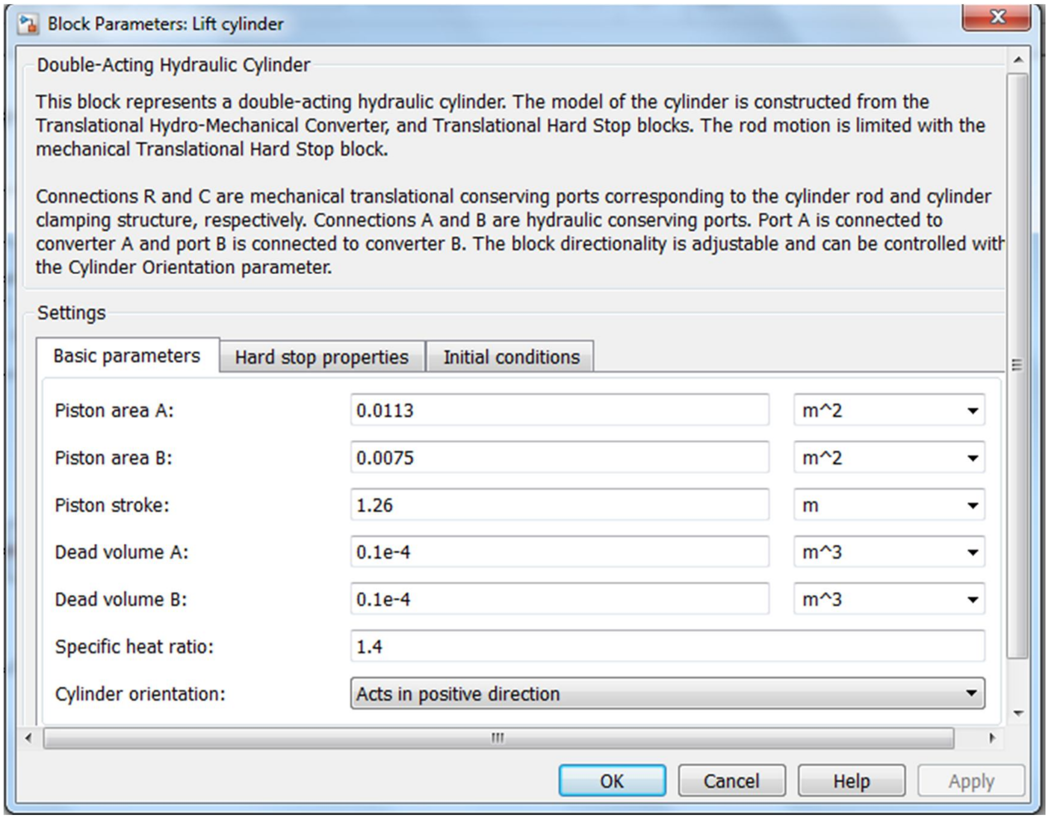

Figure 3.13: Parameters in Double-Acting Hydraulic Cylinder (Simple) block

3.7.2 Excavator Load Model

The load Weight is modeled using an ideal source of mechanical energy that generates force proportional to the input physical signal and also it contains a gain block that takes into account the gravity acceleration and conversion of the mass Boom. The source is ideal in a sense that it is assumed to be powerful enough to maintain specified force at its output regardless of the velocity at source terminals (see figure 3.14 and figure 3.15).

Connections R and C, ideal source of mechanical energy, are mechanical translational conserving ports. Port S is a physical signal port, through which the control signal that drives the source is applied. Connection C is connected to the hydraulic cylinder rod. You can use the entire variety of Simulink® signal sources to generate the desired force variation profile. Positive signal at port S generates force acting from C to R. The force generated by the source is directly proportional to the signal at the control port S [9]. The TF (Translational Friction) block represents friction in contact between moving bodies, which acts as a friction fore generated in the system.

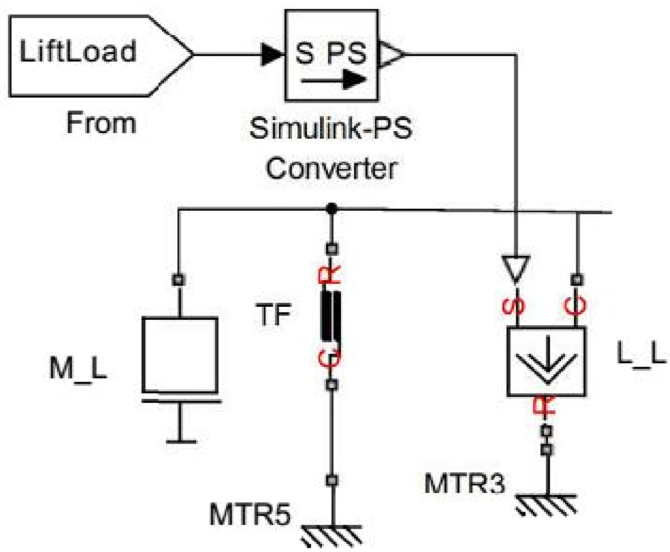

Figure 3.14: Load System

We can see in figure (3.15) the full excavator subsystem model:

3.8 Control Module Lift Load

The Signal Builder block allows us to create a profile of load variation through the time task, see figure 3.15.

3.9 Sensors Model

There are three types of sensors used in the model:

- Position sensor

- Pressure sensor
- Flow rate sensor

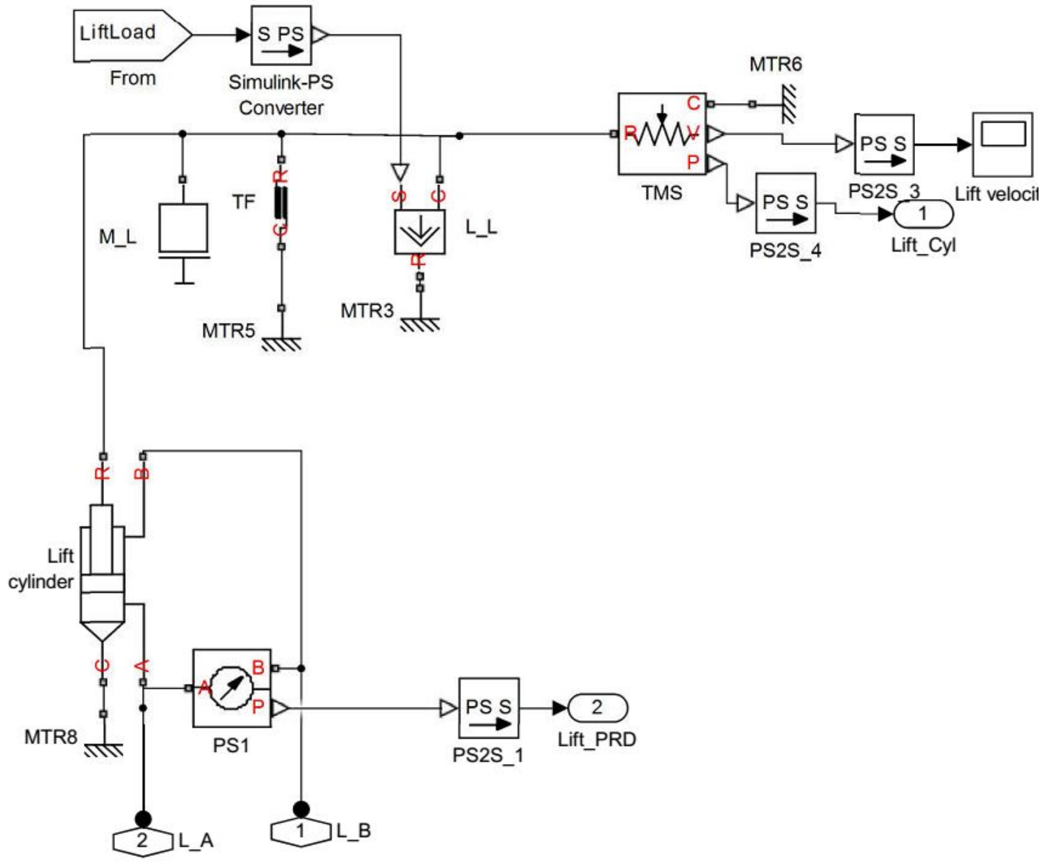

Figure 3.15: Full Excavator Subsystem

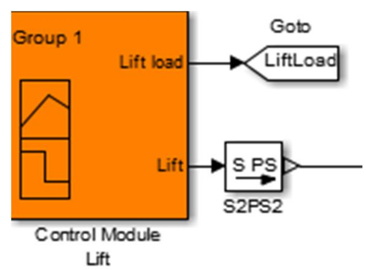

Figure 3.16: Control Module Lift Block

3.9.1 Position sensor

The position sensor is modeled using the position sensor blocks as illustrated in figure 3.17

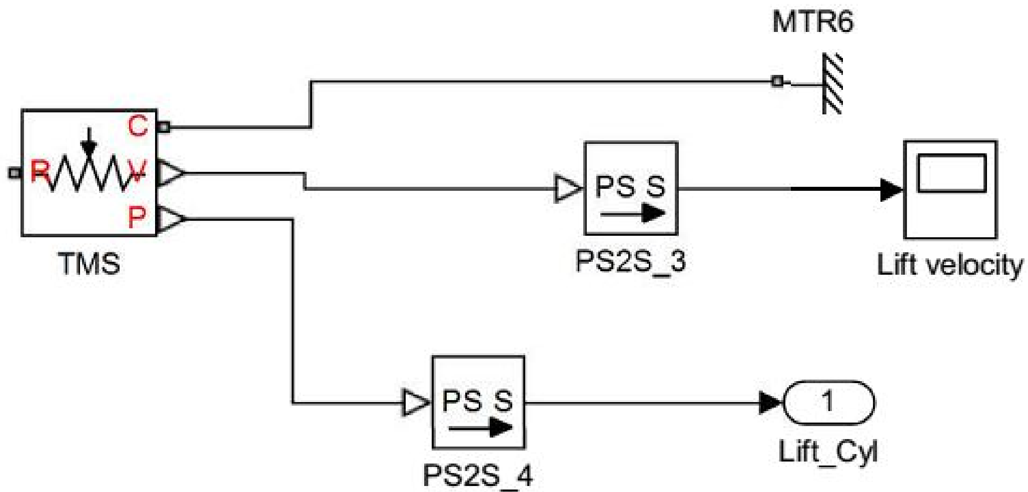

Figure 3.17: Position Sensor

The Ideal Translational Motion Sensor block represents a device that converts an across variable measured between two mechanical translational nodes into a control signal proportional to velocity or position. You can specify the initial position (offset) as a block parameter.

Connections R and C are mechanical translational conserving ports that connect the block to the nodes whose motion is being monitored. Connections V and P are physical signal ports that output the velocity and position values, respectively, for display [9].

3.9.2 Pressure Sensor

The pressure sensor is modeled using the pressure sensor blocks as illustrated in figure 3.18.

The Hydraulic Pressure Sensor block represents an ideal hydraulic pressure sensor, that is, a device that converts hydraulic pressure differential measured between two points into a control signal proportional to this pressure. The sensor is ideal because it does not account for inertia, friction, delays, pressure loss, and so on [9].

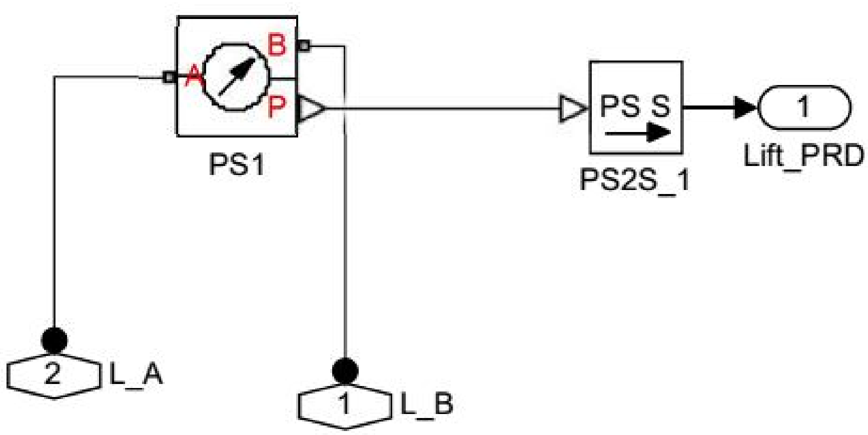

Figure 3.18: Pressure Sensor

Connections A and B are conserving hydraulic ports connecting the sensor to the hydraulic line. Connection P is a physical signal port that outputs the pressure value for display. The sensor positive direction is from A to B.

3.9.3 Flow Rate Sensor

The two flow rate sensors were modeled using the flow sensor block as illustrated in figure 3.19.

The Hydraulic Flow Rate Sensor block represents an ideal flow meter, that is, a device that converts volumetric flow rate through a hydraulic line into a control signal proportional to this flow rate. The sensor is ideal because it does not account for inertia, friction, delays, pressure loss, and so on [9].

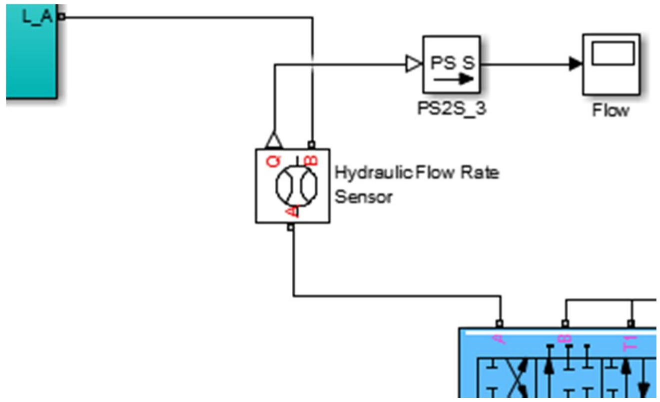

Figure 3.19: Flow Sensor

Connections A and B are conserving hydraulic ports connecting the sensor to the hydraulic line. Connection Q is a physical signal port that outputs the flow rate value for display. The sensor positive direction is from A to B. This means that the flow rate is positive if it flows from A to B [9].

VERIFICATION OF THE MODEL

In Chapter three, hydraulic control system model is developed. The objective of this chapter is to verify the model experimentally. For verification, measured variables on the real system will be compared with simulation results.

In modeling and simulation environment, all system variables can be measured. However the situation is different for real world. In this study measurements are done directly on the simulation the measured variables are:

1. Raising and lowering time in seconds.
2. Oil pressure of raising and lowering in Kpa.
3. Oil flow rate in Liter/minute.

The simulation will be compared against the required values which are available in the data book.

4.2. Measuring Instruments and Data Acquisition

As mentioned in Chapter two the hydraulic control system is equipped a position sensor and pressure sensor which are the variables to be measured.

4.3. Model Verification

As in chapter three the motion of Excavator has Tow Simulink Models Group1 Boom Raising high speed with two Pumps. Group2 Boom Normal Raising and Lowering with one Pump.

In two Groups the simulation models found in etch Simulink. The simulation results is the cylinder raise against time, cylinder pressure against time and flow rate against time and compared this simulations with the real values.

4.3.1 Group 1

In this group was used two Pumps and two main valves which is connected with two control modules so there was two same flow rates and two same pump pressures and two same cylinder pressures according to the Raising (high speed). When removed the joystick to let the Boom raise the control module in signal builder get simulation results plots, the Lift load against time and the other Lift against time (see figure 4.1).

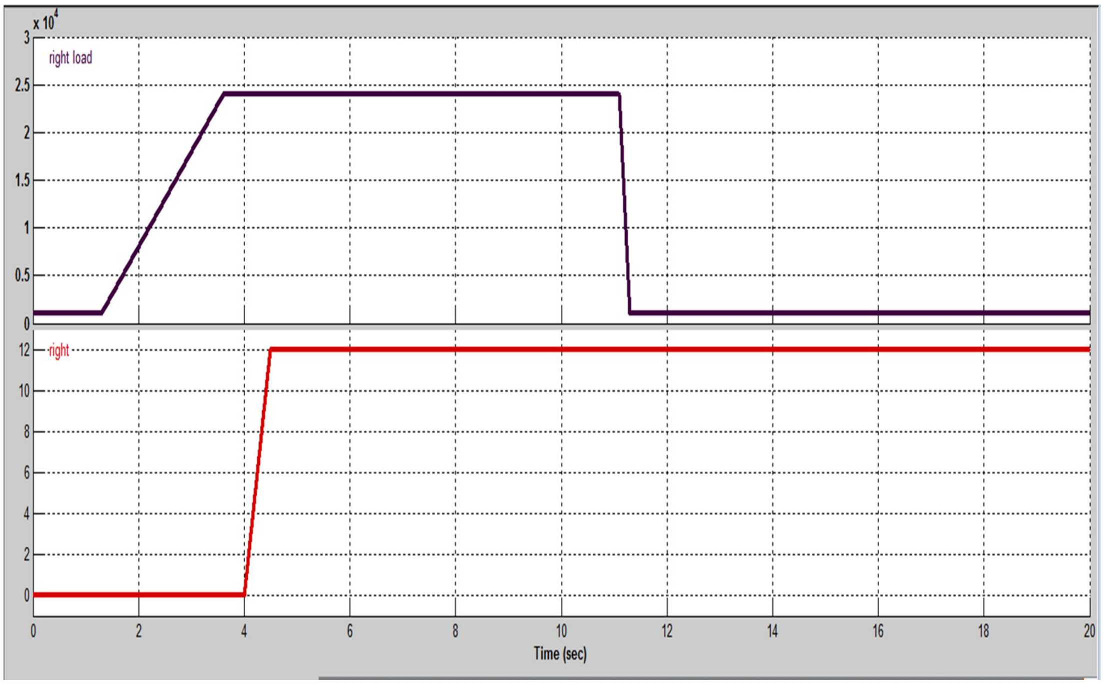

Figure 4.1: Control Module Lift- Boom Raising (High Speed)

In boom raise the simulation model will run automatically according to the parameters of the Raising in high speed. the simulation result plot the pressure cylinder against time (see figure 4.2) in this Simulation Y axes is the pressure cylinder (kpa) and X axes is the time (Sec). the pressure in 0 up to t= 1.5sec and it raise in two sec up to 21kpa and it constant from t= 3.5sec until t= 9sec in t=4.5sec overshoot pressure, and its raise again up to maximum pressure 86kpa and constant in the level up to end **.**

In figure 4.3 the simulation is boom lift against time the Y axes is lift (m) , and X axes is the time . in t=4.4sec no raise and it raise in this time at t=4.4sec up to maximum lift 1.26m and constant in these level up end.

In figure 4.4 the simulation result flow rate against time , the Y axes is the flow rate (L/min) ; and X axes is time . its in initial point up to t= 4.2sec and it increase to maximum flow rate 196L/min about 4 sec and decrease after that to 0 again until to end .

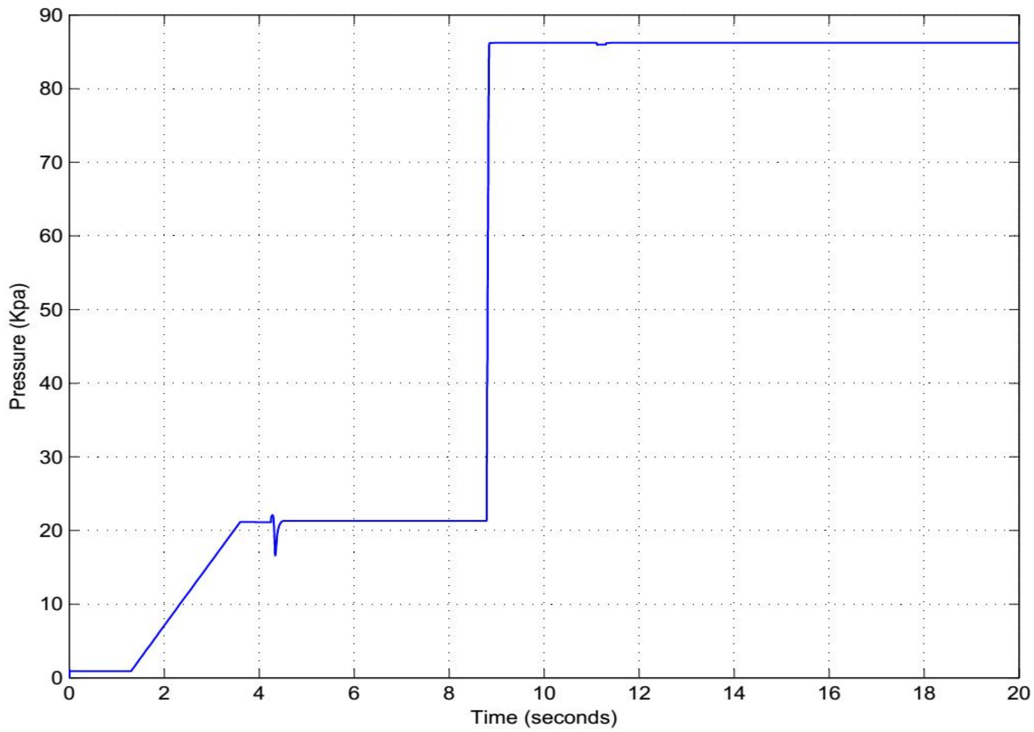

Figure 4.2: Simulation Result Plot for Pressure in Boom Raise (H. Speed)

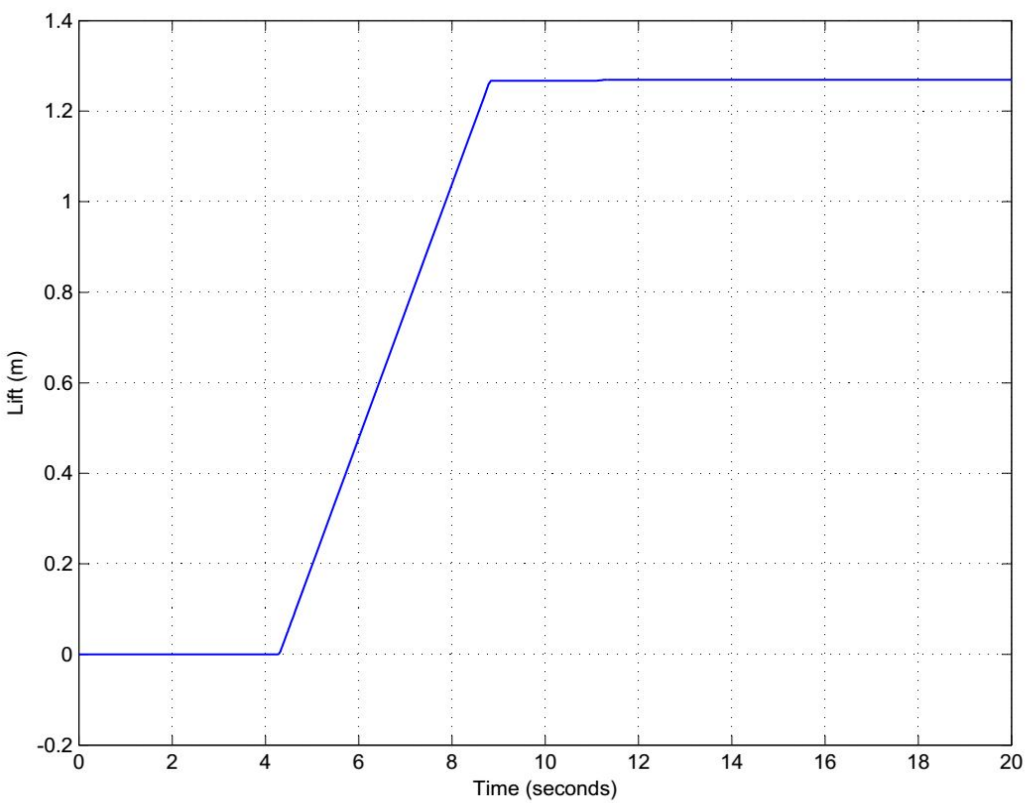

Figure 4.3: Simulation Result Plot for Boom Raising (High Speed)

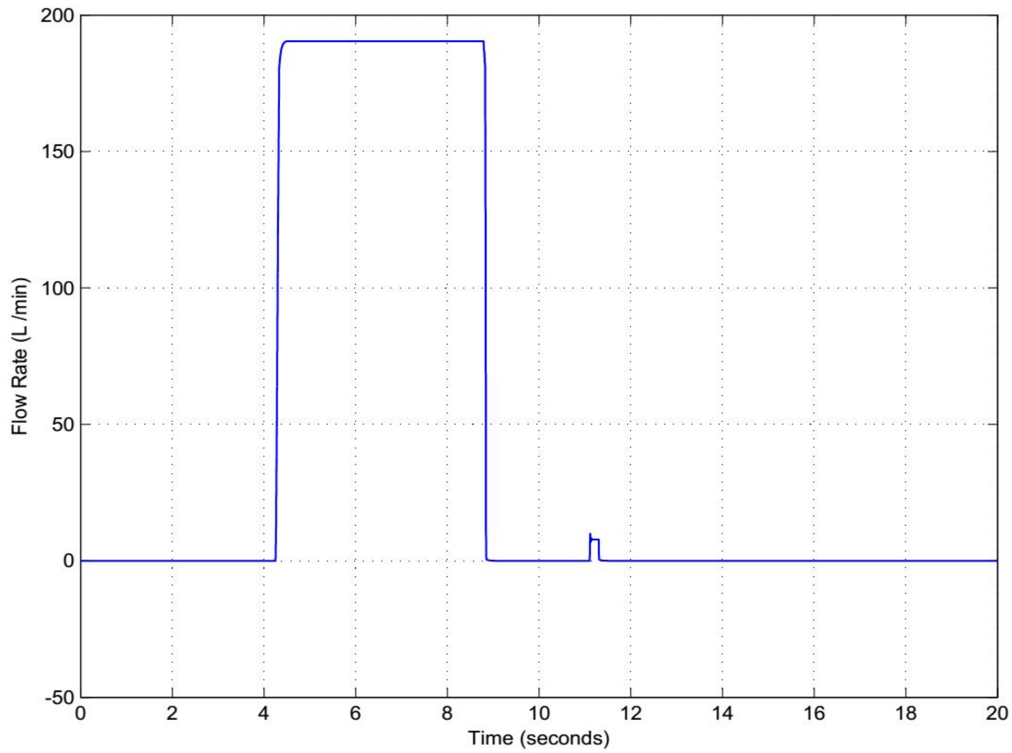

Figure 4.4: Simulation Result Plot for Flow in Boom Raising (H. Speed)

Table 4.1 shows the comparison between the required values against simulation . The flow rate error; the cylinder pressure error and boom mover error was found from the results of Boom raising (high speed) the error in boom mover is 33% , the error in Cylinder pressure is 40% and the error in Oil flow rate is 1% see Table 4.1.

Simulation results was relatively acceptable compared to the required values. It can be noticed that there is overshooting in the cylinder pressure at t=4.4sec this overshoot could not be noticed on the real system . may be due to low response time of the pressure the cause of this undershooting is due to the sudden stop of the cylinder piston after it reached the bottom end of the cylinder. that is why the maximum error in cylinder pressure.

RESULT	SIM	REQUIRE VALUES	ERROR
TIM(Sec)	4.2	2.8±0.5	33%
CYLINDER PRESSURE (kpa)	96	58±2	40%
OIL FLOW (L/min)	196	194±2	1%

Table 4.1: Comparison Between the Required Values and Simulation

4.3.2 Group 2

In this group used only the right pump and right main valve. There was two motions so two control module lifts which pilot the main valve in raise we can connect the control module lift and in lower we connect the control module lift lower this two motions in normal speed .

When removed the Joystick to get the Boom raise "Normal Raising" with one pump first in signal builder the simulation results plots Lift load against time and Lift against time low (see figure 4.5).

In boom raise the simulation model will run automatically according to the parameters of the Normal Raising simulation result plot the pressure cylinder and pump pressure against time (figure 4.6) the Y axes is the pressure (cylinder & pump) the X axes is a time in cylinder pressure is raised after t=1.5sec and there is overshoot in t=4.4sec and constant up to t=11sec and decrease after that to 0pressure up to t=13.2sec and increase to maximum pressure up to end, pump

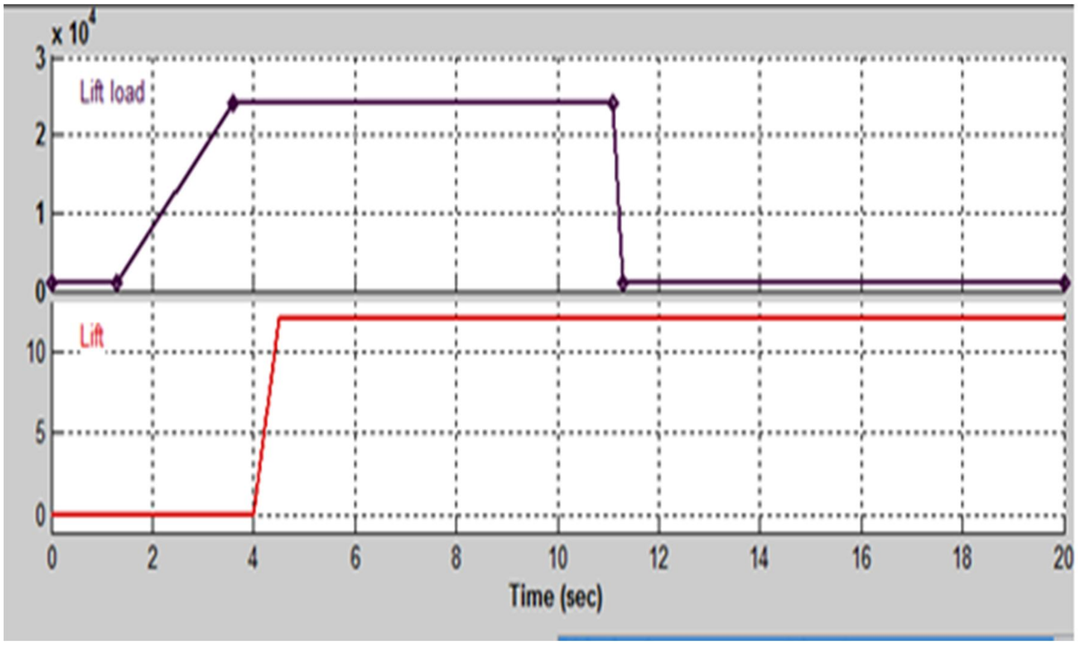

Figure4.5: Control module lift-Boom Normal Raising One pump

t=11.2sec the pressure is 36kpa and decrease after that up to pressure =17kpa and constant up to t=13.2 and identical with the cylinder pressure.

In (figure 4.7) the simulation result plots the raise cylinder against time, in t=4.4sec it raise up to lift=1.26m in t=13.2sec and constant up to end.

In figure 4.8 the simulation result plots the flow rate against time the Y axes is the flow rate and the X axes is the time the flow is initial point up to t=4.2sec and increases to maximum flow rate =95L/min up to t=13.4sec and decrease after that up to initial point at t=13.4sec constant up to end.

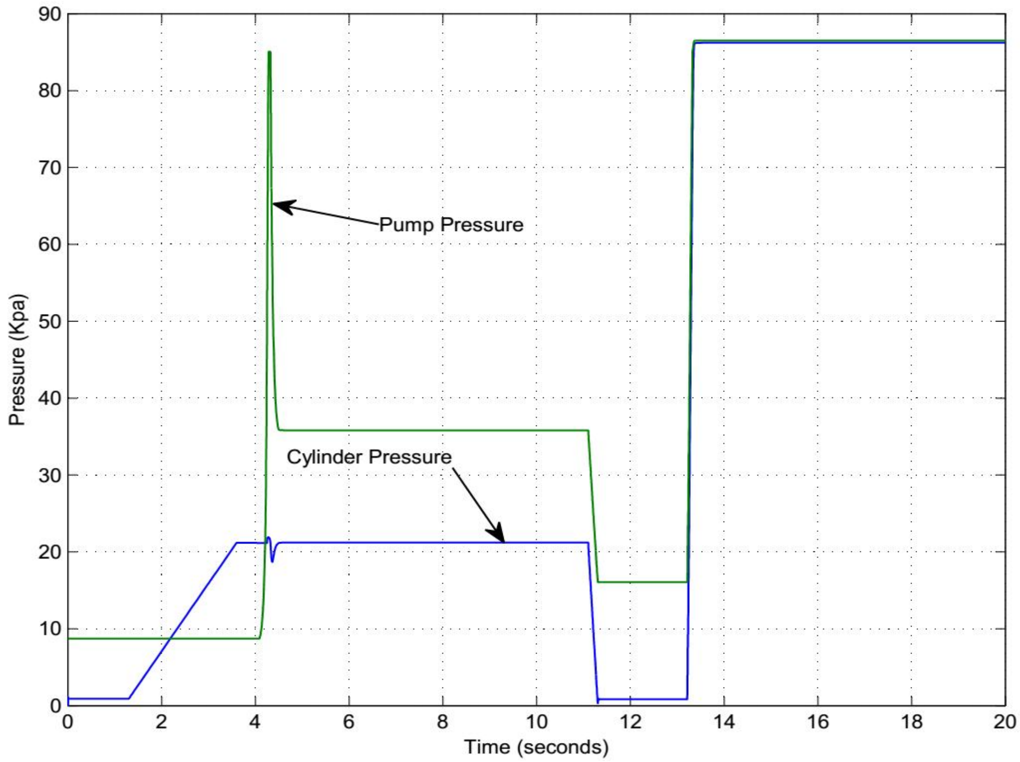

Figure 4.6: Simulation Result Plots for P. Cylinder And P. Pump(N. S)

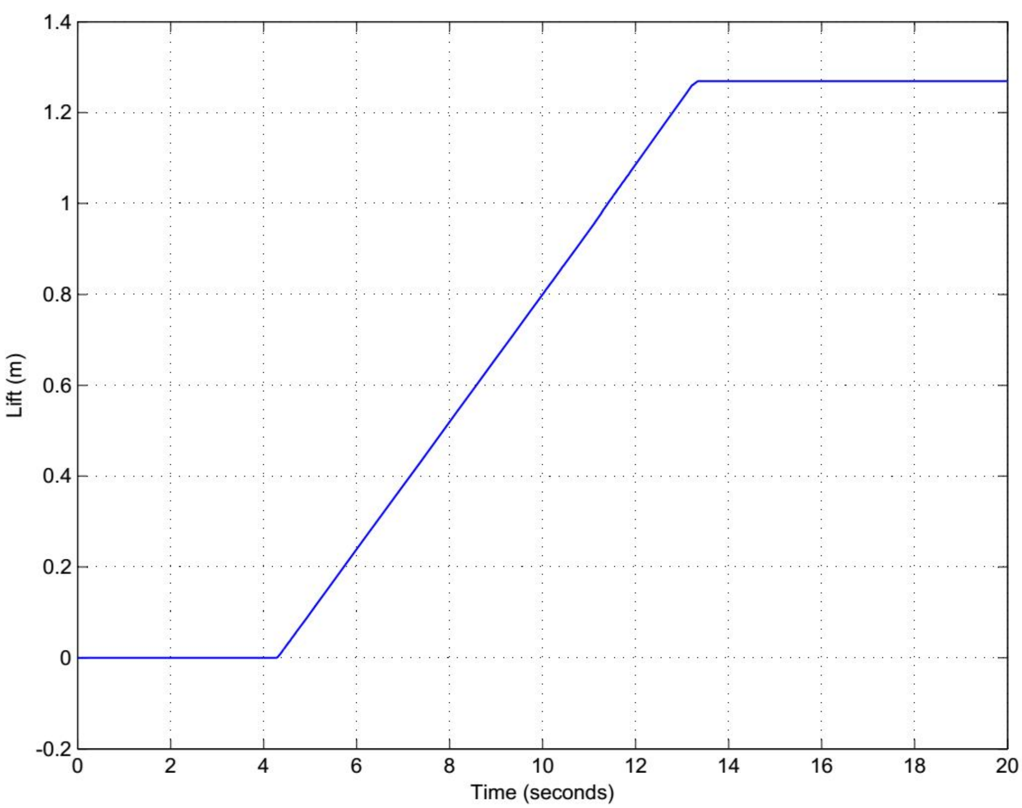

Figure 4.7: Simulation Result Plot for Boom Normal Raising One Pump

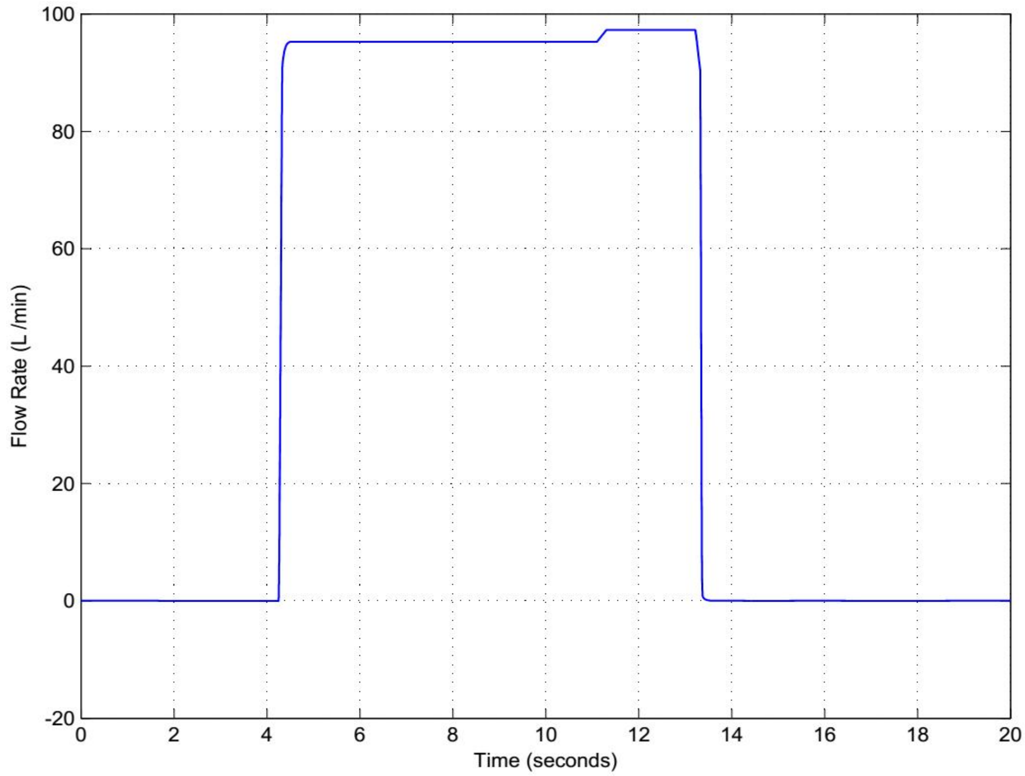

Figure 4.8: Simulation Result Plot For Flow Rate (Boom N. Raising)

When removed the joystick to gets the Boom lower (Normal Lowering) with one pump. first in signal builder gets two simulations result plots; the Lift load against time and the Load against time (see figure 4.9) .

In boom lower the simulation model will run automatically according to the parameters of the Normal Lowering. Simulation result plot the pressure cylinder and pump pressure against time (see figure 4.10) the Y axes is the cylinder pressure and pump pressure and the X axes is the time the cylinder pressure is initial point up to t=1.5sec and increase up to pressure =21kpa in t=3.5sec and overshoot in t=4.4sec and constant up to t=10.2sec and increase to negative in t=11.2sec its negative pressure up to end, the pump pressure =9kpa from t=0sec up to 4.2sec and overshoot and pressure =36kpa up to t=10sec and overshoot and decrease to 9kpa and variable up to end.

In Simulation result plots the lower cylinder against time (figure 4.11) the cylinder in 0m up to t=4.4sec and raising up to maximum lift=0.84m in t=10.3sec and it constant up to t=13.6sec and decrease to .05m and constant up to end.

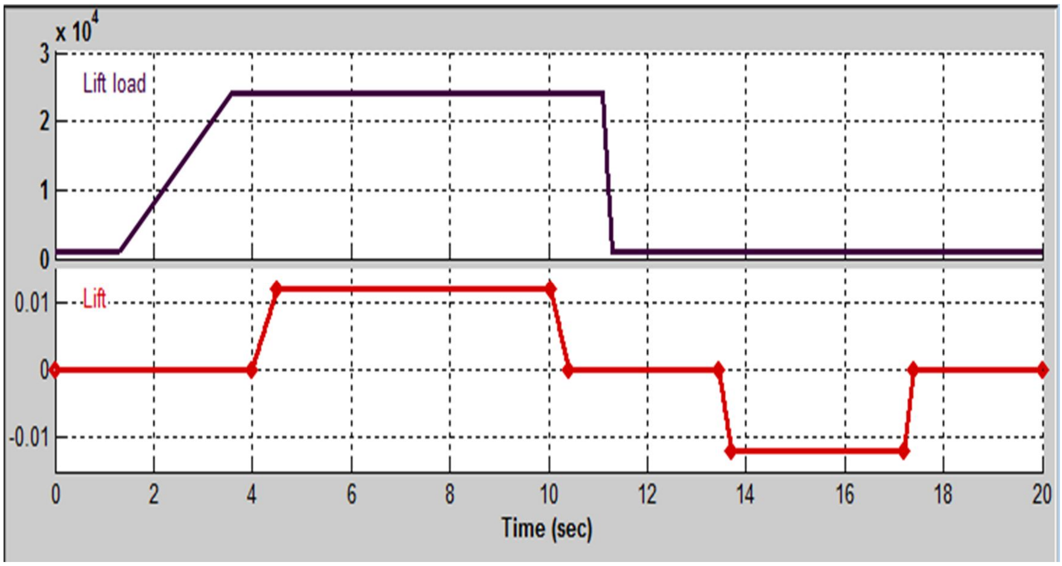

Figure 4.9 : Control Module Lift- Boom Normal Lowering One Pump

In Simulation result plots the flow rate against time (figure 4.12) the Y axes is the flow rate and X axes is the time the flow rate is zero up to t=4.2sec and it increase up to 95L/min in t=4.2sec up to t=10.2sec and it decrease to zero again up to t=13.6sec and it lowered to -145 up to t=17sec and raise to zero up to end.

Table 4.2 shows the comparison between the required values against simulation . The flow rate error, the cylinder pressure error and boom mover error and the pump pressure error was found from the results of boom raising and boom lowering (normal speed). In normal raising the error in boom mover is 15% , the error in Cylinder pressure is 16%, the error in oil flow rate is 2% and the error in pump pressure is 6% . In normal lowering the error in boom mover is 35% , the error in oil flow is 2%, the error in cylinder pressure is 16% and the error in pump pressure is 6% see Table 4.2

Simulation results was relatively acceptable compared to the required values. The maximum error is 35% in boom mover in normal lowering . the reason is that the weight is biasing in lowering in require value so the weight decrease the lowering time but in simulation the weight have no biasing .

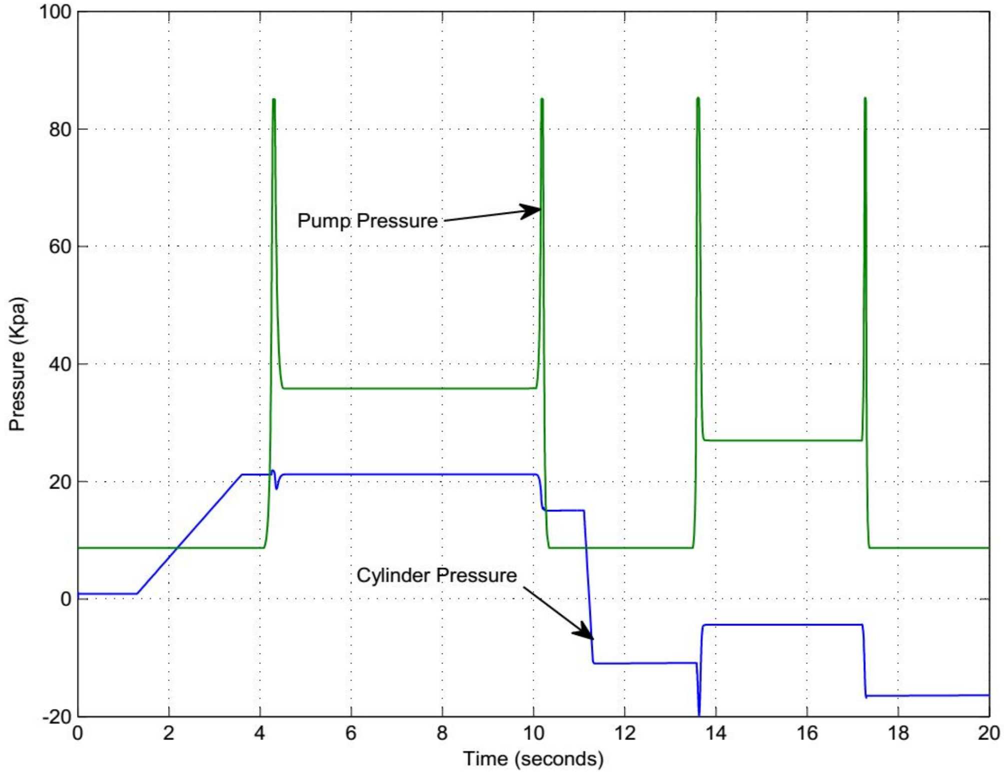

Figure 4.10:Simulation Results Plots For Pressures and Pump Pressure (N. Lower)

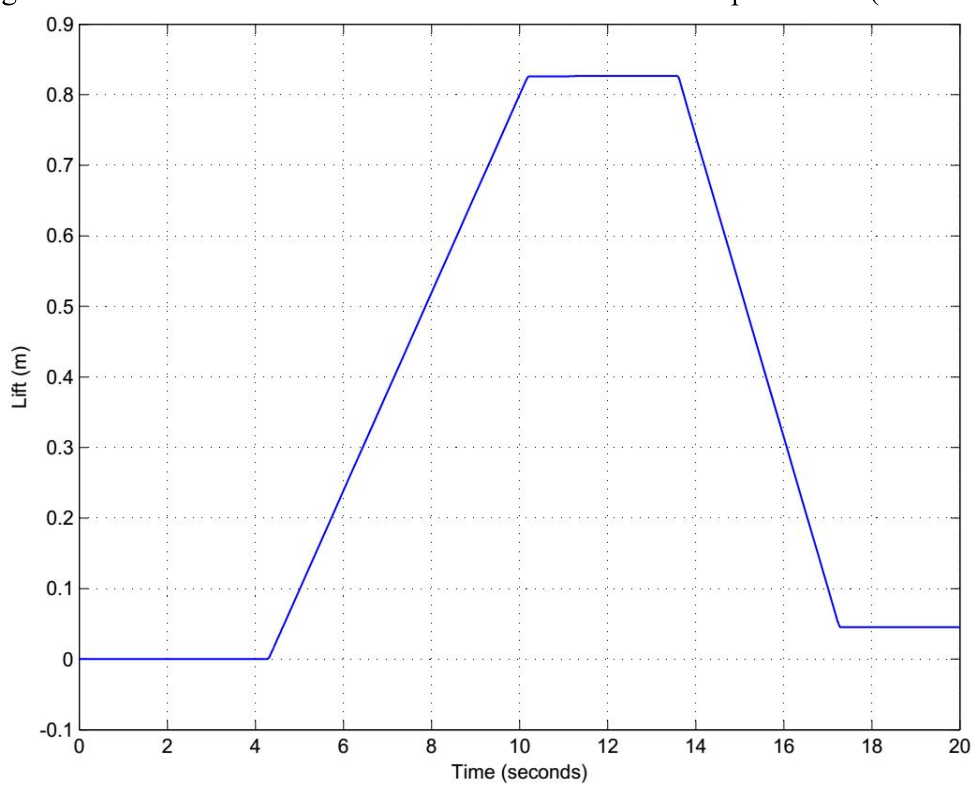

Figure 4.11: Simulation Result Plot for Boom Normal Lowering One Pump

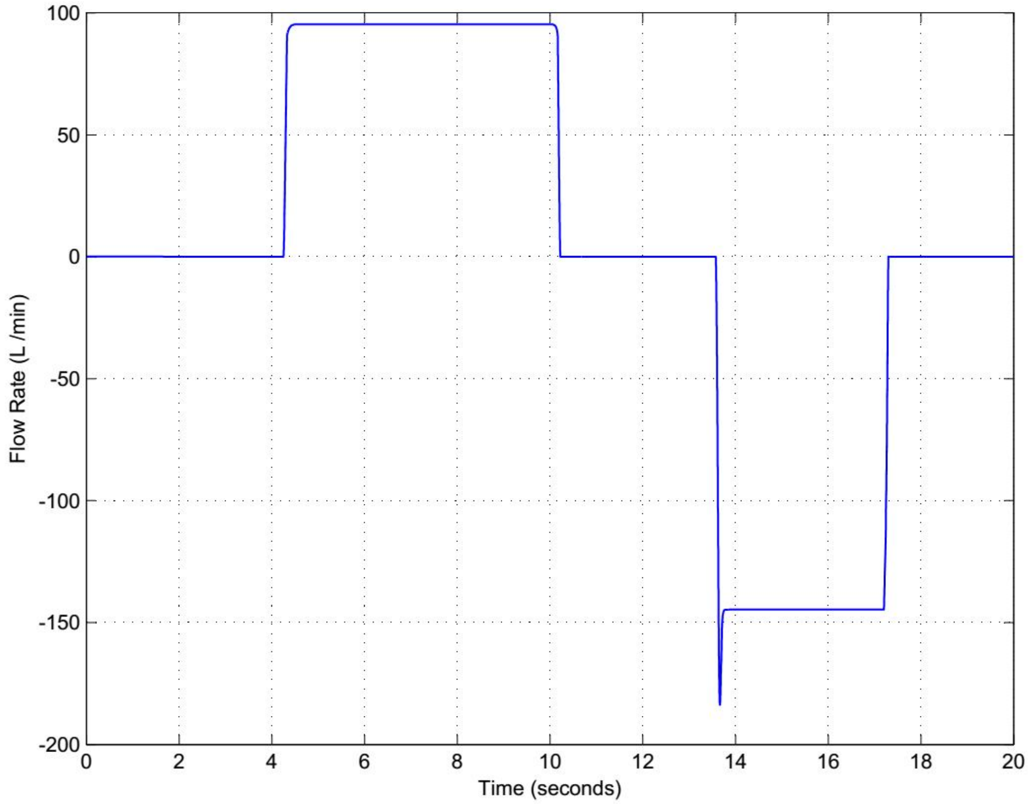

Figure 4.12: Simulation Result Plot for Flow Rate (Boom Normal Lowering) One Pump

	NORMAL RAISING			NORMAL LOWERING		
RESULTS	SIM	REAL VALUES	ERROR	SIM	REAL VALUES	ERROR
TIME (SEC)	5.9	5.6 ± 0.5	15%	5.9	3.8 ± 0.5	35%
OIL FLOW (L/MIN)	95	97 ± 2	2%	95	97 ± 2	2%
CYLINDER PRESSURE (Kpa)	21	25 ± 2	16%	21	25 ± 2	16%
PUMP PRESSURE (Kpa)	36	34 ± 2	6%	36	34 ± 2	6%

Table 4.2: Comparison Between the Required Values and Simulation

CONCLUSION AND RECOMMENDATION

5.1Conclusion:

In this research focuses on modeling of a hydraulic control system of excavator motion dynamic. MATLAB/Simulink® software is used to develop the model.

Instead of deriving the hydraulic system equations , physical toolboxes inside MATLAB environment are to model the hydraulic of the machine hydraulic system response due to the dynamic motion of the rigid parts obtained by co-operating the hydraulic simulation in Simulink plat form. Standard blocks in SimHydraulics® library are used for modeling hydraulic pumps, hydraulic fluid, check, relief, pressure and directional valves.

The values from the data book is Compared with simulation results, which is recorded from the Simulink model. In addition, the model is utilized in order to analyze the relation between the boom motion and raising time.

Cushioning Block was not included in the model, since there is no available data for it and trial and error method followed many times to find its parameters but resulted in suspending MATLAB software many times.

In conclusion, a MATLAB/Simulink® model for hydraulic control system is developed in this work. Prepared model is verified by comparison of simulation and real values. The developed model can be used as base model and mechanical model for the boom movement can be integrated to it. Finally, points that are open to improvement can be determined over the model and changes in the system parameters can be simulated before implementation on the real system.

5.2 Recommendations for Future Work

In future can modeled simulation of the Stick hydraulic system and the Bucket hydraulic system and also can modeled the dynamic simulation and calculate the forces on the bodies . The result of that models can be ability to determine the dynamic loads on the joint and attachment of the

Excavator . in addition to that prototyping time and cost can be highly reduced by implementing that model in the design process.

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